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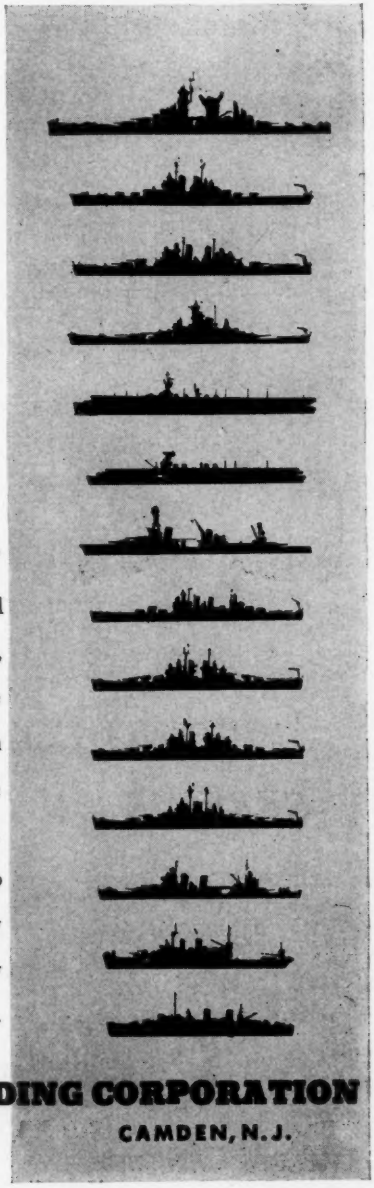

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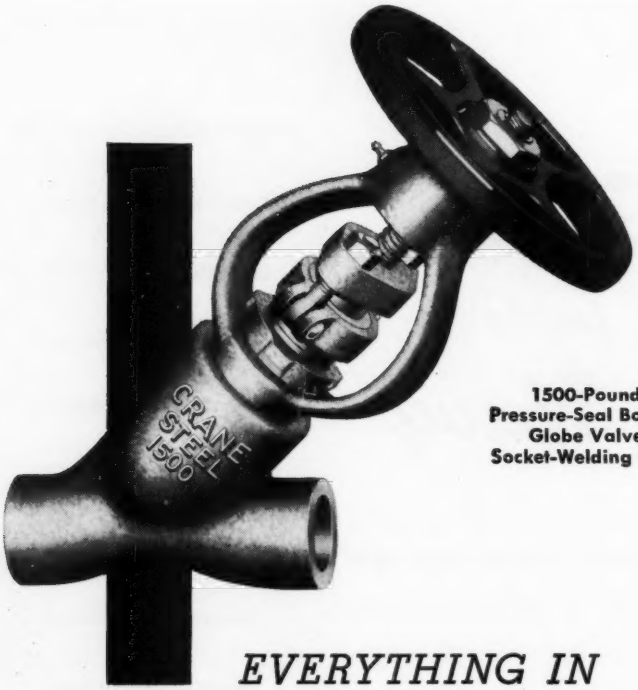

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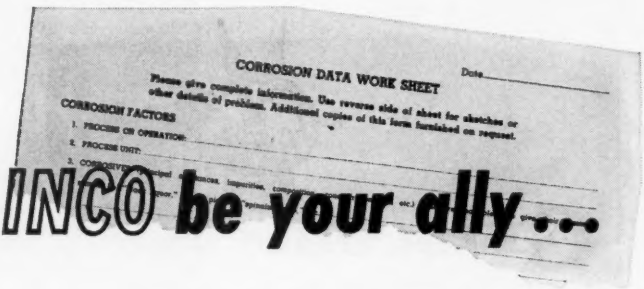
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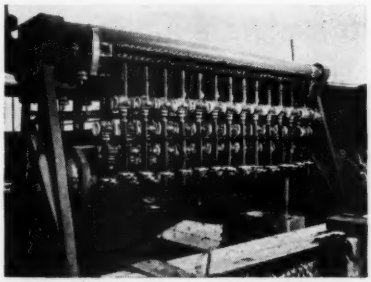

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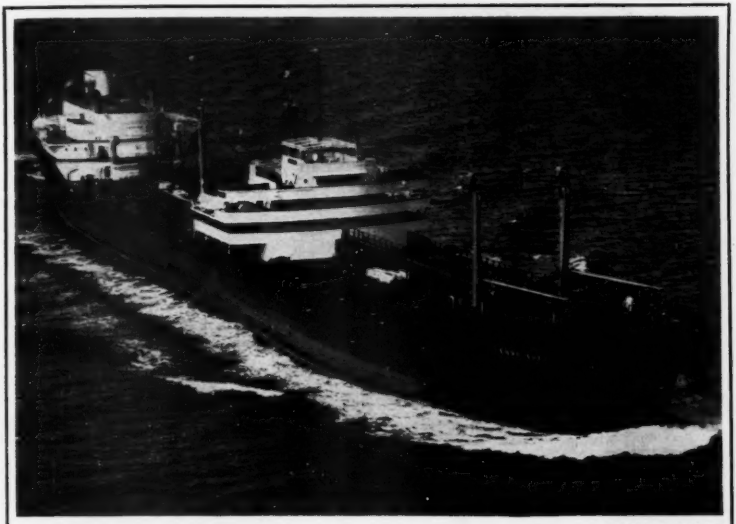

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TANKER PROGRESS

The eleventh, and last, of a group of modern 628-foot tankers of the above type has now been delivered by the Newport News Shipbuilding and Dry Dock Company. With a capacity seventy per cent greater than the World War II T-2 tanker, these tankers travel ten per cent faster at a speed of sixteen knots.

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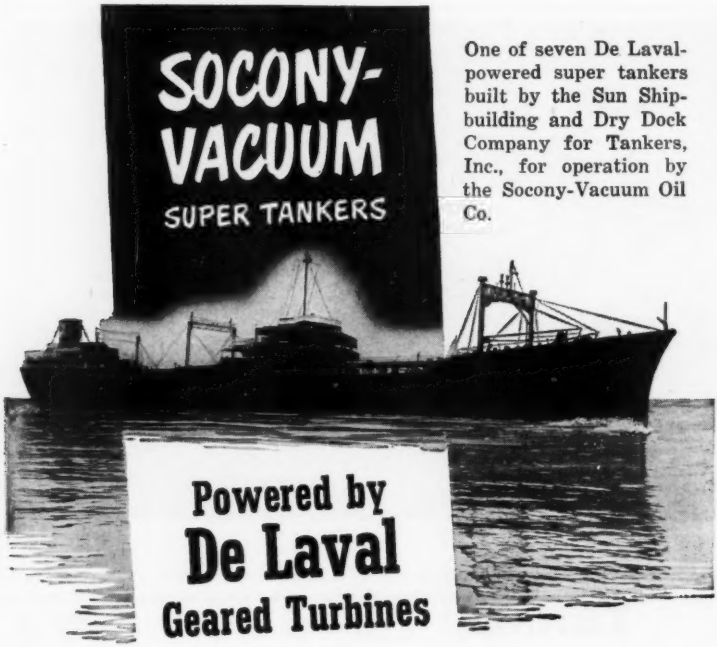

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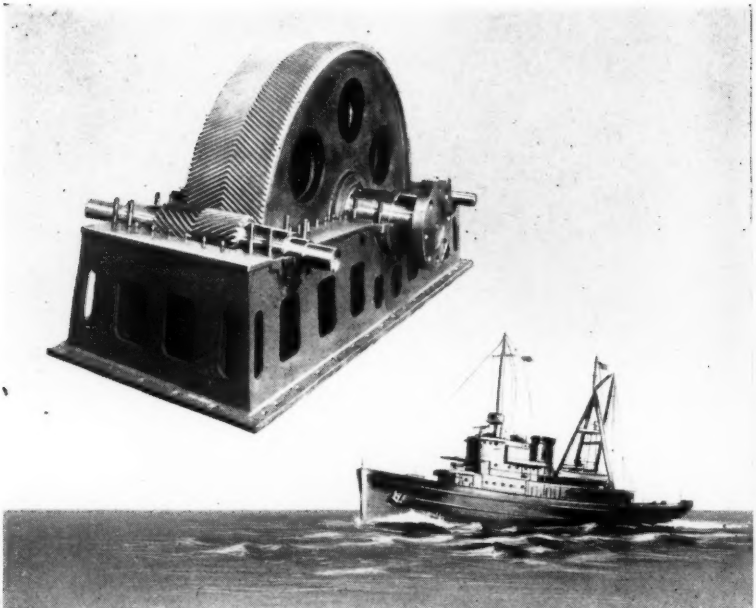

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The American Merchant Marine— Mid-Century Analysis

THE AMERICAN MERCHANT MARINE CONFERENCE sponsored by The Propeller Club of the United States in conjunction with its Twenty-fourth Annual Convention, will be held in Baltimore, Md., September 27th, 28th and 29th, 1950. For the first time since 1941, this joint meeting will be held outside of New York.

The Conference theme will be "The American Merchant Marine — Mid-Century Analysis." Truly, in this year 1950 the American Merchant Marine is confronted by an intensely serious situation. Problems of vital concern must be solved if our merchant shipping is to serve effectively as a bulwark of defense and as a medium for the transportation of our domestic waterborne commerce and a substantial portion of our foreign trade.

Panel Discussion Meetings will be held on September 27, 28 and 29th; the joint Convention-Conference Luncheon on Thursday, September 28th; Propeller Club Convention Session on Friday, September 29th; and on the evening of that date, the annual American Merchant Marine Banquet. At all of these sessions, outstanding leaders representing government and the industry will present their thoughts on subjects of the utmost importance affecting the future of the American Merchant Marine.

Today, more than ever before, constructive and coordinated efforts and action are essential . . . The cooperation and participation of everyone possessing an interest in the American Merchant Marine are needed . . . Arrange now to attend the American Merchant Marine Conference and take part in this very important gathering of the American Marine Industry!

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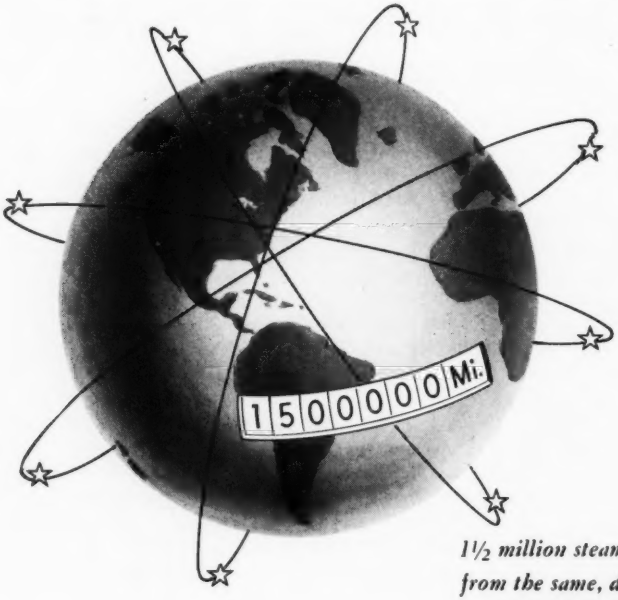

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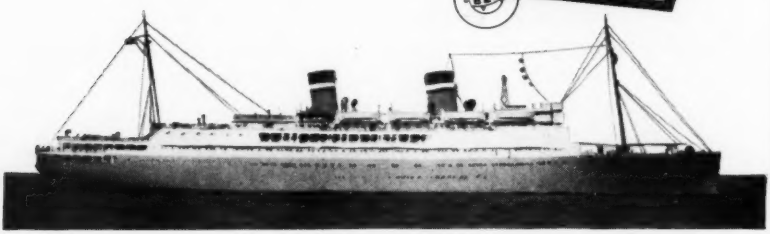

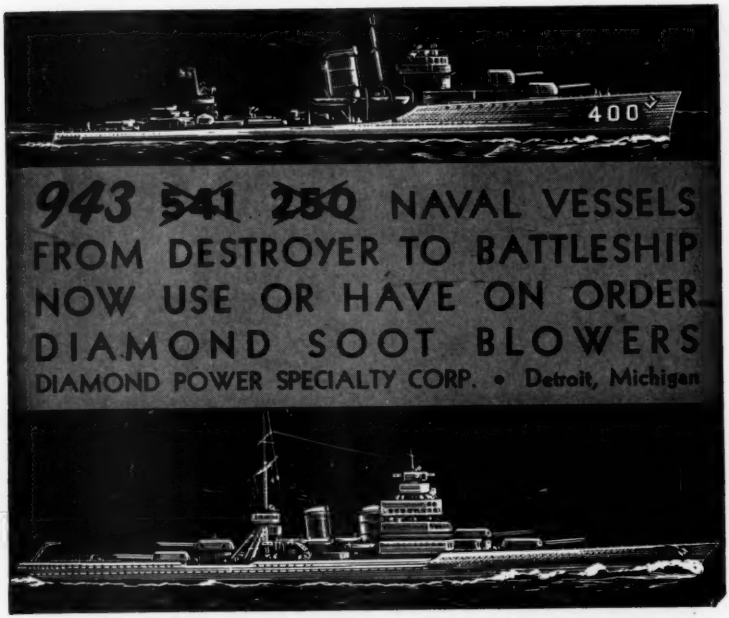
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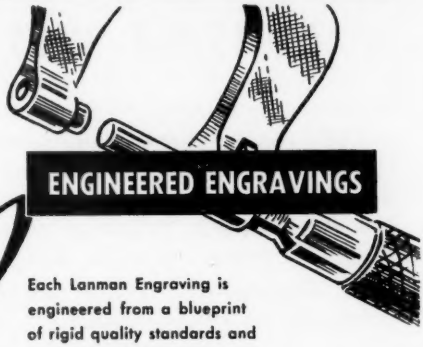

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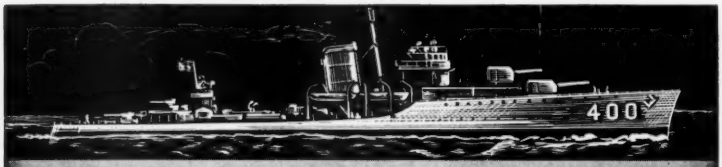

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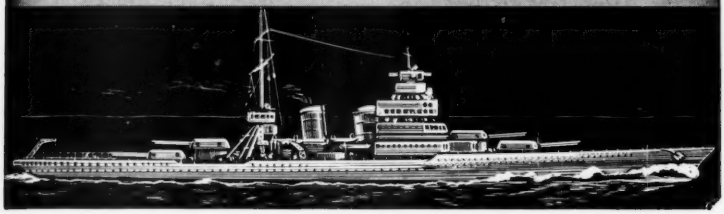

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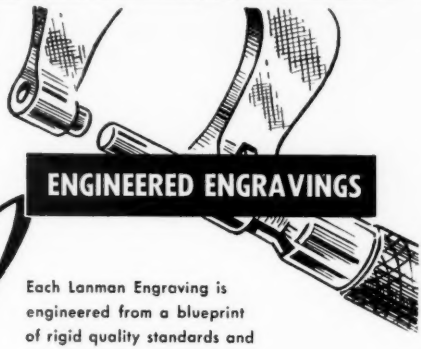

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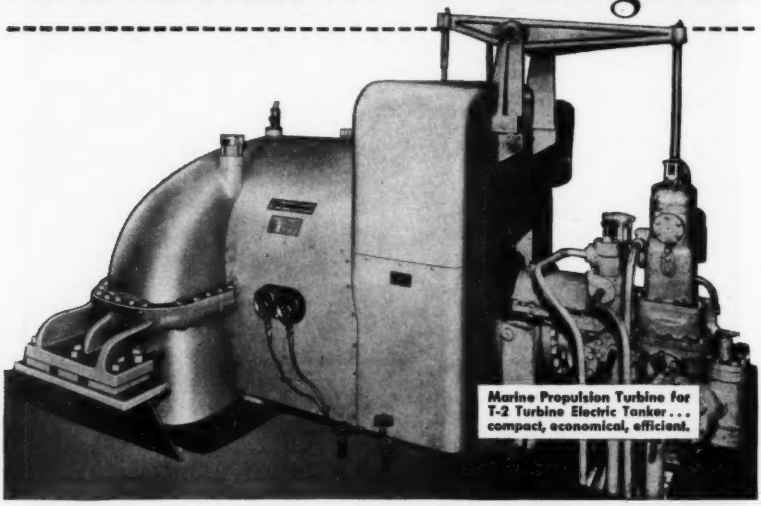

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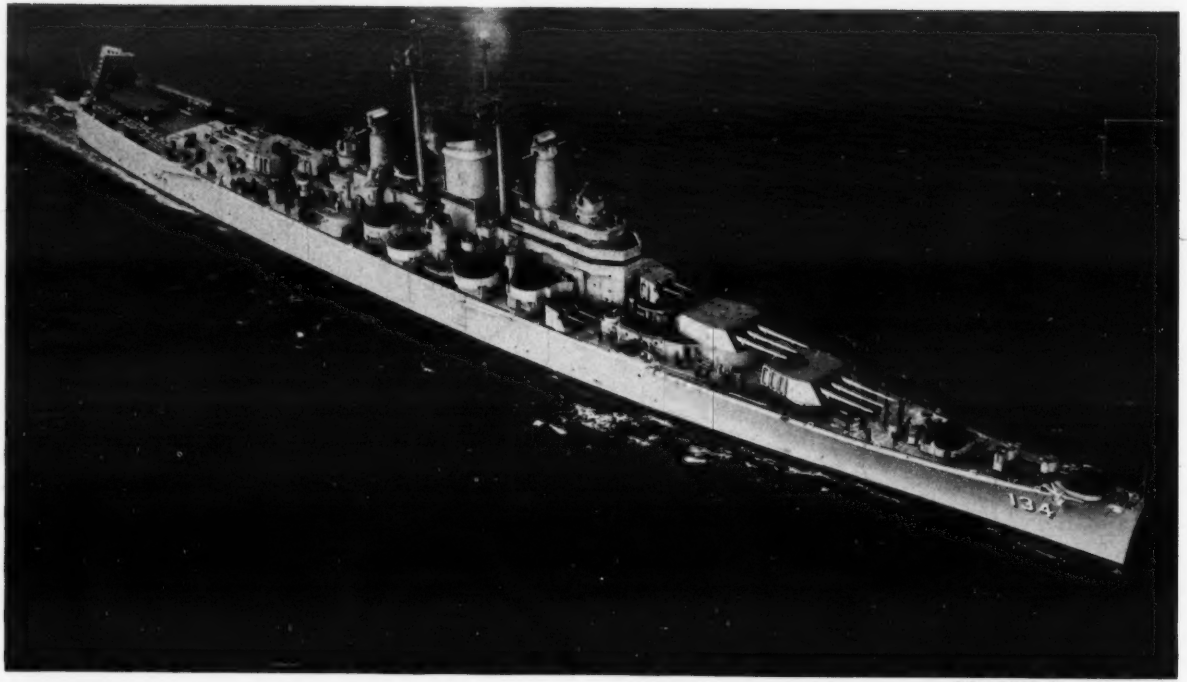

Official U.S. Navy Photograph

USS *Des Moines*

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JOURNAL

OF

THE AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.

VOL. 62. AUGUST 1950. No. 3

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SECRETARY'S NOTES

CHANGE OF ADDRESS

Effective at once the mailing address of the Society is changed from the Navy Department which is the address used for the last 62 years to:

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The Washington telephone number, until further notice will remain unchanged. It is COlumbia 5289.

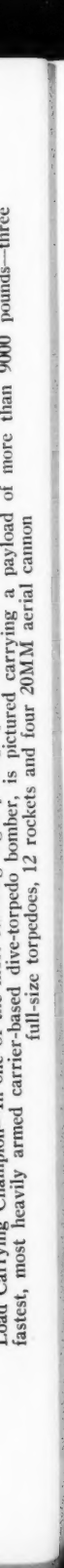
LIST OF MEMBERS

The list of members as of 1 May 1950 was mailed during June. Members who are interested in maintaining correct mailing lists may do so by using that list and "Changes in Membership" which appears in this and subsequent issues of the JOURNAL.

USS Des Moines

Official U.S. Navy Photograph

Load Carrying Champion—In one of the most startling flight photographs ever made, a Martin AM-1 Hauler, the Navy's largest, fastest, most heavily armed carrier-based dive-torpedo bomber, is pictured carrying a payload of more than 9000 pounds—three full-size torpedoes, 12 rockets and four 20MM aerial cannon



Load Carrying Champion—An one-place [illegible] bomber, is pictured carrying a payload of more than 9000 pounds—three full-size torpedoes, 12 rockets and four 20MM aerial cannon

fastest, most heavily armed carrier-based dive-torpedo

PART ONE

ORIGINAL ARTICLES

The articles which the Editor, acting under the supervision of the Council, has accepted for publication in Part One are the original works of the authors.

They are offered in furtherance of the Society's raison d'etre: to further the advancement of naval engineering. Discussion of any article is invited for publication in a subsequent issue.

Permission to reprint is freely granted on condition that *full acknowledgment is given* to the author and to the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS.

All articles which have been written by naval personnel and are published herein carry the personal view of the authors and, unless specifically so stated, do not express the official views of the Navy Department.

Official U.S. Navy Photograph

Largest Test Gun Fires 2,000-Pound Bombs and Guided Missile Warheads at Supersonic Speeds

INTEGRATED SPECIFICATION PRODUCT PERFORMANCE

in

Two Parts

PART I. THE PROBLEM—ELLSWORTH F. SEAMAN

PART II. THE CAPACITOR—PHILIP D. BELL

THE AUTHORS

ELSWORTH F. SEAMAN studied Electrical Engineering at the Carnegie Institute of Technology, Pittsburgh, Pennsylvania. After leaving school in 1923 his professional career between 1923 and 1931 consisted of engineering work with electrical utility companies in Pennsylvania, New York, Alabama and Texas. In 1931 he entered the employ of the Bureau of Engineering. During his nineteen years' service with the Navy Department he has been in electrical engineering work in the research, development and standardization categories. He is currently acting in the Navy Department civilian administrative capacity for the Armed Services Electro-Standards Agency, Fort Monmouth, New Jersey, and is engaged in general standardization planning work for the Bureau of Ships, including International Standardization.

PHILIP D. BELL graduated from Pennsylvania State College with a degree in Electrical Engineering in 1937. From this time until 1942 he was employed in engineering work with electrical manufacturers in Pennsylvania and Ohio. Joining the staff of the U. S. Navy in 1942, his time has been spent mainly in directing test activities at Naval Research Laboratory and Material Laboratory. Associated with the Bureau of Ships from 1 January 1948, his responsibilities have been divided among three electronic component activities: research and development, preparation of military specifications, technical administration of government and non-government laboratory testing.

Largest Test Gun Fires 2,000-Pound Bombs and Guided Missile Warheads at Supersonic Speeds

Integrated Specification Product Performance

Part I

The Problem

The science of semantics has provided man with a method of thought communication through the spoken or written word, but a challenge to the adequacy of this means of thought transfer arises in the evolution and use of a purchase specification. Starting with the premise that we are a part of a literate environment it is certain that combinations of words in specification form will always convey meanings to the producer that may or may not be consistent with the interpretation intended by the consumer and specification writer.

The hazard of ambiguity is present in all types of specifications (design or formulation and performance*) and it is therefore essential that the finished document be tested by the questions: Does any statement in the specification have more than one interpretation? Have the test methods for measuring performance characteristics been accurately defined and are they reproducible? Has the intent of the consumer to get the most for his money been frustrated by giving a poor or mediocre product the edge over a superior product? Each of these questions is in itself the basis for lengthy discussion but this paper is primarily concerned with the last and with the availability of ready and workable answers.

There is considerable leeway in the handling of business matters in industry, for the exercise of engineering judgment in the acceptance or rejection of a bid on a product covered by a specification. There are many cases where a product falls below the specified limit for a relatively minor characteristic but exceeds the minimum requirement by a wide margin on an important performance factor. The obvious thing would appear to be acceptance of the better product having the best overall performance but with slight departures in some respects from the specification requirements; such action can be taken in non-government procurement. If, however, this practice were followed during government purchase it would immediately give rise to the question of discrimination, or, expressed in another form; how could all competitive bidders have the same chance to compete if all of the rules for acceptable bids were not made known?

The most obvious corrective step is to readvertise under revised specifications that lower the unimportant limit and raise the important ones, if in making this latter change it is still possible to get adequate competition and availability. This practice, however, has many drawbacks, one of which is the time consuming procedure of rewriting a specification and readvertising. It also poses another problem of how much a minimum limit can be lowered before it is under the level of adequate performance. There is no good answer to the man who points out that his product will pass a dielectric stress test of 1390 volts per mil compared to a specified minimum of 1400 v/m, so why not lower the minimum requirement since a decrease of 0.7 percent is not really significant? The contention may be true but the ultimate conclusion of successive reductions is a gradual approach to zero.

It is the aspect of quality that makes evaluation of a product difficult by conventional performance specification technique. It is known from engineering experience that exceptionally high performance for one characteristic usually

*Performance specifications are defined as those requirements that specify what is needed by the consumer in terms of end use performance. Design or formulation specifications tell the producer how to make the product by specifying design data and formulae.

results in the compensatory lowering of performance for another factor. The engineer tries to allow for this by predicting where the different levels of performance will occur and then reflecting his conclusions in minimum specification requirements. A difficult problem occurs when the products of different producers all have the lower performance values occurring for different characteristics. One type of magnet wire may have improved heat resistance but less ability to remain concentric within the insulation than a second type where the reverse is true. The acceptance of minimum performance values taken from a summarization of all competitive products can result only in general reduction of quality.

The difficulty arises out of the inflexible nature of the conventional type of specification that is available for use. The criterion here is that the product must qualify as many times as there are characteristics to be met; it is not the summation of performance of the decathlon where an individual win or near win will accumulate point credits commensurable with competitive skill shown, but where ultimate victory requires the greatest show of total competence regardless of individual successes or failures. Assuming that the decathlon principle or the integrated total performance is applied to the specification how could it be worked out so as to be administratively practical, non-discriminatory in character and technically sound by acceptance of the good and exclusion of the poor?

It was suggested* that the work of Walter B. Pitkin in his book "Psychology of Achievement" might furnish the clue that would lead to the answer. In this work the author analyzes the careers of various types of people from moron to genius. Each characteristic of the man contributing to success or failure is plotted in magnitude on one of the spokes of a wheel emanating from a central hub. When the parameter is analyzed for the completed curve it is remarkably coordinated in the genius circle for personages like Paderewski and Edison; for others of lesser calibre, parameters of lower magnitude occur. In using this principle for industrial products it would be impracticable and unrealistic industrywise, to require the genius level of performance and unsatisfactory from the service reliability standpoint to drop below the median level. Excluding the exceptional or non-standard cases, the preferred strata of potential supply should fall in the vicinity of level II, Fig. 1, where dots represent sources of supply.

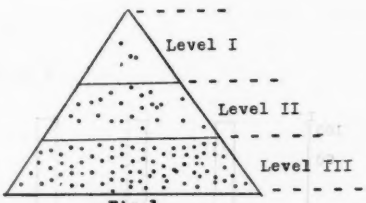


Fig.1

A study of these ideas has developed one technique of evaluation that follows the basic concept of integrated evaluation; it is set forth as Part II. The more complicated integrated area of the wheel spoke arrangement has been eliminated in favor of the bar graph type of computation. The number of samples has been equitably considered by reducing performance of all samples to an equivalent single sample. Points are credited to the score for each test that is made and the total score is called the "Performance Factor."† The performance factor value is arbitrarily established based on engineering knowledge, consumer needs, and commercial availability of the product. The factor is a

*Suggested by Mr. J. N. Hall, Electronics Engineer, Bureau of Ships, Navy Department.

†This should be distinguished from some so-calfed "work factor" types of specifications where credit is given for extra high performance but where a fractional departure from specification limits means rejection.

valid one if it does not reject obviously good components or accept poor ones.

In using this approach to acceptability of a product under a performance specification it is important to establish certain rules for its application. It is consistent with the philosophy of this discussion that performance factor technique should apply only to product quality and not to factors involving minimum functional requirements or dimensional interchangeability needs. These two latter categories involve usability of a product and should be measured by the "go" "no go" limits. If a relay coil does not operate at the right "pick-up" voltage or if the mounting dimensions do not fit the space available then the product cannot be used. These categories are therefore not within the scope of performance factor evaluation.

It is obvious after study that any system of evaluation will involve a question of relative importance of different performance characteristics and the assignment of proper weights. One solution to this and other problems is proposed in Part II using the capacitor as a specific product example.

Integrated Specification Product Performance

Part II

The Capacitor

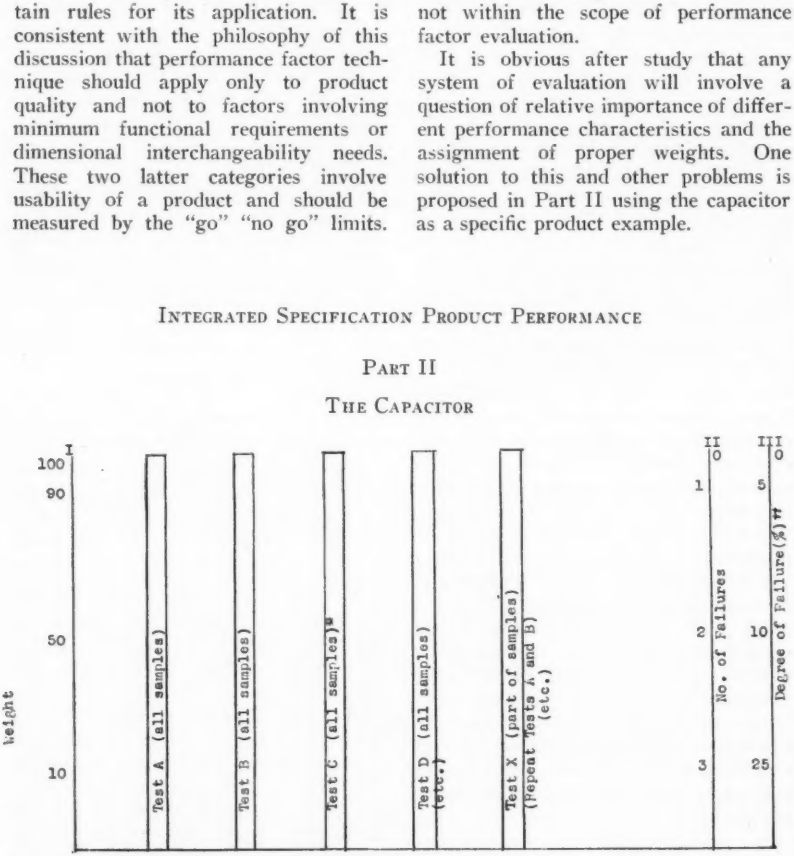


Fig. 2.

*Not applicable to "Degree of Failure".
**No more than 3 samples shall fail any test.
††If "Degree of Failure" is over 25%, score zero.

(1) Weight for each test is derived from the sum of: (a) Where failures occur, from proper point on Col. II carry horizontally to Col. I for weight. (b) Where failures occur, from proper point on Col. III carry horizontally to Col. I for Weight.

(2) Where two failures occur, "Degree of Failure" point on Col. II is selected from average of both failures.

(3) Total Weight for all samples on all tests in summation of all weights calculated in (1) above.

The chart illustrated by Fig. 2 represents a method of analyzing test results under electronic component specifications, offering considerable latitude not obtainable with the method now used. While this chart is general, it can be adapted to any particular specification, by modification of the listed tests, and number of samples tested. It should be noted that the numerical values shown in the Figure are somewhat arbitrary, but can be altered as desired.

This method judges test results not on the basis of predetermined failure tables, but on a scale of numerical failures and degree of failure for any test. For test (A), if no samples fail, a score of 200 is attained, 100 each for "No. of Failures" and "Degree of Failure." If one sample were to fail by being 12.5% over the nominal rating where 10% is permitted, then a score of 90 is given for "No. of Failures" plus a score of 10 for "Degree of Failure," or a total of 100 for test (A) out of a possible maximum of 200. If two samples fail by both being out of tolerance by 11% where 10% is permitted, then the score for this test would be 50 for "No. of Failures" plus 50 for "Degree of Failure."

In some tests indicated by an asterisk in Fig. 2, there is no "Degree of Failure"; the test is a "go—no go" proposition. Dielectric strength is a good example of this. In such cases then a score of 200 is assigned on the basis of "No. of Failures" only. There are still other types of tests which do not lend themselves to this proposed method:

SUMMARY OF TESTS. TYPE CM35E562K

Test	*Par. No. of JAN-C-5*	*No. Tested*	*No. Failed*	*No. Failures Permitted*
As received				
Visual	F2c	48	0	0
Dielectric Strength	d	48	0	1
Insulation Resistance	e	48	0	0
Q	f(1)	48	0	0
Dissipation Factor	f(2)	48	0	0
Temperature Coeff.	h	6	0	0
Capacitance Drift	h	6	0	0
Capacitance	i	48	1	0
After Thermal Cycle & Immer.	j			1
Dielectric Strength	d	12	0	
Insulation Resistance	e	12	0	
Capacitance	i	12	0	
After Vibration & Corrosion	k			2
Visual	c	12	0	
Dielectric Strength	d	12	1	
Insulation Resistance	e	11	0	
Capacitance	i	11	0	
Life	l			1
Visual	c	12	0	
Dielectric Strength	d	12	1	
Insulation Resistance	e	11	0	
Capacitance	i	11	1	

"Visual" is such a test. In these cases the "Visual," "Salt Spray" and similar tests be judged by the test engineer as presently done, with no attempt being made to write these tests into the score making up an integrated performance factor.

To best appraise the value of this proposed method, let us examine the present method along with the proposed against typical test results. The following table represents a summary of tests of type CM35E562K capacitors, with ratings of 5600 mmfd. ± 10%, 500-VDCW, ± 100 ppm/°C temperature coefficient, ± 0.3% capacitance drift. Armed Service specification JAN-C-5 has been selected for purposes of study. According to Table IV of JAN-C-5, these samples have failed to qualify and must be rejected. Details of the samples which failed are as follows:

(a) One of 48 samples failed to be within its capacitance tolerance, measuring 6250 mmfd. (with measurement error applied favorably to the measurement), as against the maximum permissible 6200 mmfd. It failed, therefore, by being almost 1% more than the allowed ± 10% tolerance.

(b) After the Vibration and Corrosion test, one of twelve samples broke down as twice rated voltage was being applied.

(c) After the life test one sample measured 6810 mmfd. whereas it measured 6050 mmfd. before the test; it changed capacitance 12.5%

In summary then, these samples failed to meet the requirements of JAN-C-5 as follows: The failing samples not destroyed by the dielectric strength test changed their characteristics by either a small or large enough amount to fall below performance requirements.

From these comments, let us proceed to analyze the data of the above table and in accordance with Fig. 3. For those tests and measurements where no failures occurred, the weight would be:

Test	
As Received	
Dielectric Strength	200
Insulation Resistance	200
Q	200
Dissipation Factor	200
Temperature Coefficient	200
Capacitance Drift	200
After Thermal Cycle and Immersion	
Dielectric Strength	200
Insulation Resistance	200
Capacitance	200
After Vibration and Corrosion	
Insulation Resistance	200
Capacitance	200
After Life	
Insulation Resistance	200
	2400

Note that if all samples pass all tests, a maximum weight of 3200 can be scored. Now, examining the data for samples that failed we have:

	No. of Failures	*Weight Degree of Failure*	
As Received			
Capacitance	80	90	
After Vibration and Corrosion			
Dielectric Strength	160	—	
After Life			
Dielectric Strength	160	—	
Capacitance	80	10	
	480	100	Total 580

En toto then, the weight of tests of these samples is 2980 of a possible maximum of 3200. We have set an arbitrary figure of 2900 as a minimum acceptable level, and therefore these samples would be accepted. This conclusion would not be the same as that reached in applying current JAN-C-5 requirements, as the basis for rejection is obviously not the same. As the simplest example of the advantages of Fig. 3 is the following:

(a) Under JAN-C-5, if there are no failures in Dielectric Strength and one failure in Capacitance (of 48 samples "As Received"), the submission fails.

(b) Under Figure 3, with the same conditions as (a), the submission need not fail pending results of other tests.

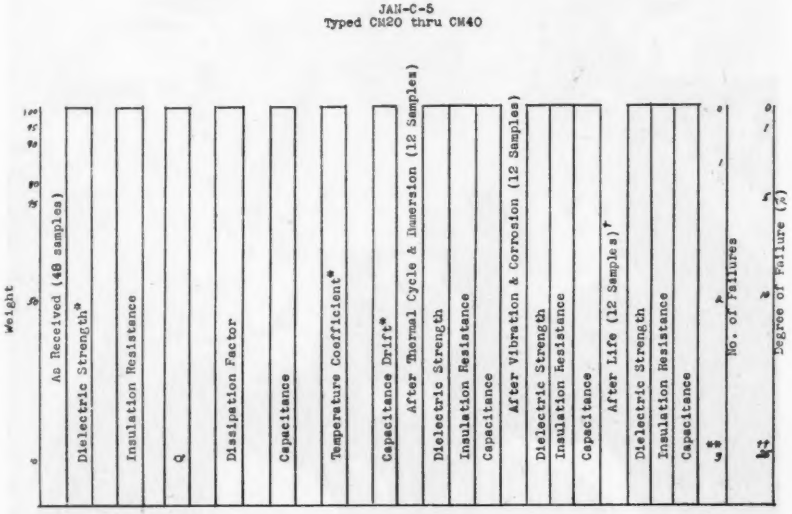


FIG. 3.

*Not applicable to "Degree of Failure".
†Samples failing in the course of the life test shall be considered, for analytical purposes, as having failed "Dielectric Strength".
**Not more than 3 samples shall fail any test.
††If % is more than 25, Weight is 0.

Official Department of Defense Photo

LSU's coming into LSD during recent exercise.

A STATISTICAL APPROACH TO THE LONGITUDINAL STRENGTH DESIGN OF SHIPS

"A Good Ship is a Safe Ship"

NORMAN H. JASPER

THE AUTHOR

The author received his Bachelor's degree in Mechanical Engineering from the College of the City of New York in 1941. He joined the Scientific section of the Puget Sound Naval Shipyard in 1941 and worked there until transferred to the David Taylor Model Basin in 1946 where he has primarily been concerned with applying vibration theory and experimental methods to the study of ship structures. He studied, since 1941 at the University of Washington, the George Washington University, the Graduate School of the Bureau of Standards and the University of Maryland from which he received the Master of Science degree in 1950.

It is the purpose of this paper to examine our present position with regard to the problem of the longitudinal strength of ships at sea and to propose a program for a study of these strength requirements. Two basically different approaches to the problem are discussed. It is suggested that a statistical study of sea conditions and of the strength requirements of ships at sea be made. A brief description of some types of instruments that will secure and in some cases analyze the required data is given. A proposal is made for a design procedure which utilizes the method of the electrical analogy.

The strength design of ships which has passed through a gradual process of evolution is still basically founded on empirical rules. The calculation of the longitudinal strength of a ship, if it is calculated at all, is usually carried out by assuming the ship to be poised on some "standard" wave. This process is fairly satisfactory for comparing various ship designs, but cannot be considered a realistic representation of actual sea conditions. There are numerous cases on record where ships have shown evidence of lack of longitudinal strength.

Information as to the actual demands made upon the hull girder when sub-

jected to the action of the sea is relatively scarce, and such information as is available is not sufficient or of a form in which it can be used in re-evaluating the present method of ship design. To date, weaknesses in ship strength have too often become apparent only after failures have occurred.

The commercial transportation field is a competitive one. It is apparent that savings in dead-weight tonnage of the vessel realized through more equitable distribution of structural material in accordance with actual strength requirements will make for more economical operation of merchant vessels by virtue of reduced power requirements, greater mileage travelled in a given time as well as through savings accrued due to decrease in the time and funds required for maintenance and shipyard time incurred through structural breakdowns.

Ship design is a highly important though complex engineering problem. Experimental and analytic approaches to design problems are coming into wider use than ever heretofore, especially in the field of airplane and machine design, where dynamic loads, high speeds and light weight are of considerable importance. It is believed that the time is at hand where shipbuilders must find a more realistic approach to the strength design of the hull girder. We must learn what demands the sea makes upon the ship girder and how these demands vary with the service to which the vessel is to be put. When such information is available, we must be able to apply it to the more equitable design of the hull grider. *A definite overall program should be arrived at which is feasible, economical of time and cost and which preferably fits into the existing pattern of ship design.*

Historically, perhaps one of the most influential analytical approaches to the determination of the longitudinal bending strength of ships was given by Sir E. J. Reed before the Royal Society in 1872. In his theory the water is supposed to assume its wave form without alteration of its hydrostatic pressure; the vessel considered is placed so as to be in static equilibrium on the crest or the hollow of a wave of length equal to that of the ship and the hog and sag moments are calculated with these assumptions.

The variation in water pressure in a wave due to the orbital motion of the wave particles was given by W. E. Smith in a paper presented to the Institution of Naval Architects in 1883.

Captain Kriloff of the Russian Navy gave several papers on the analytical treatment of the stresses experienced by a ship in a seaway. His papers are excellent treatises but they do presume a knowledge of the existing waves and do require extensive calculations not suitable for general application.

In 1905 Professor J. H. Biles reported on the British full-scale tests that had been conducted on H.M.S. *Wolf* to determine the validity of the simple beam theory as applied to ships. In 1931 Commander C. O. Kell of the U. S. Navy conducted static tests, in general similar to those made on the *Wolf*, on the destroyers *Preston* and *Bruce*. Both these tests indicated that the simple beam theory can be assumed to apply for the case of static loading of ships.

In 1938 Professor G. Schnadel presented a paper to the International Conference of Naval Architects and Marine Engineers in which he reported on the sea tests of the *San Francisco* during the course of which wave configurations, deflections, accelerations and pitching angles were measured. These tests are particularly significant because the vessel encountered severe storms. One rather steep wave was about 186 meters long and 16.5 meters high, illustrating the point that a ship must be designed to withstand the most severe loading which it may encounter although it may be advisable to permit the peak stresses to exceed the yield point under the most severe loading, since such loads may

occur but rarely in the life of a vessel. The *San Francisco* test indicated the possibility of using the ship itself as a measuring apparatus in order to determine the wave conditions to which it is subjected. A later set of sea trials during which extensive measurements were taken has been discussed by the British investigators F. B. Bull and A. J. Johnson in a paper given before the Institution of Naval Architects, September 1948, entitled "The Measurement and Recordings of the Forces Acting On a Ship at Sea." It will be some time before a complete analysis of the data obtained by them will be available. The full-scale tests discussed are typical of a number of similar ones made to date.

In all the tests and investigations noted above, the vessel has been assumed to act as a rigid body, that is the vibratory motions of parts of the ship relative to each other have been neglected.

In investigations of the longitudinal strength of ships it has generally been assumed that the ship can be considered as a floating beam and deflections and stresses have been calculated on the basis of simple beam theory. Simple beam theory assumes that the beam is homogeneous throughout its length, the cross section is uniform throughout and that the deflections are small. Extensive full-scale static tests, such as were made on the destroyers *Bruce* and *Preston* show that the basic beam theory is applicable to the ship girder. However, it does not necessarily follow that the beam theory can also be applied to the dynamic behavior of ships, that is, under dynamic load application. In order to check the applicability of the simple beam theory to the analytical treatment of dynamic loading, the David Taylor Model Basin conducted tests during which loads varying sinusoidally with time, were applied to a ship by means of a "vibration generator."—

"The vibration generator is a machine which essentially comprises two rotors geared together in such a manner that they rotate in opposite direction. Each rotor carries unbalanced weights. The centrifugal forces generated by the unbalanced weights are phased so as to produce a resultant sinusoidal force. The frequency of force application may be varied by varying the rotor speed and the magnitude of the force may be changed at any given speed, by changing the effective unbalanced mass carried by each rotor."—

The frequency of load application was varied from a slow rate to as high as 1000 load cycles per minute. It was found that the hull did respond as though it were a beam for frequencies of load application up to several hundred cycles per minute. At the higher frequencies the vessel no longer behaved as an integral beam, the various components of the hull girder vibrated as individual plates, frames, etc. However, the frequencies of stress variations which are of prime importance in the strength of ships at sea are most likely those corresponding to the lower modes of vibration of the ship and for these the tests showed the vessel to behave as a beam.

The problem before us is: to design the ship girder so as to withstand the most severe loading that it will encounter throughout its intended service and to do so in a manner which results in an equitable distribution of strength throughout the vessel. The design procedure should be based on known service requirements and should be economical in time and cost.

Previous investigators have provided analytical methods for determining the strength in a seaway of a rigid hull girder which methods require, however, a knowledge of the sea conditions. These procedures have not come into general use due to their relative complexity. The method proposed by Sir E. J. Reed in 1872 is still in use today although somewhat refined. Reed's method is readily applied and does give an index

of the relative strength of ships. The full-scale static tests of Professor Biles and Commander Kell have shown the validity of the beam theory for static loading, the vibration generator tests made by the David Taylor Model Basin have shown that this assumption may reasonably be extended to the dynamic behavior of ships. The wave, strain and motion measurements made on the high seas have indicated the applicability of the simple beam theory and have shown the great complexity and variability of the waves. The latter tests are expensive to conduct and are much too few in number to provide any statistical basis for ship design in general.

It is evident, after review of accomplishments to date, that the major need today is for sufficient data to provide a statistical basis for the prediction of strength requirements for the various types and sizes of hulls operating in the several major bodies of water. It is necessary to plan the manner in which this type of data is to be applied in order to determine the quantities that will need to be measured as well as the manner in which they are to be related.

There are two principal methods of attacking the problem. The first method requires that the external forces acting on the vessel be measured and that the response of the ship to these forces be calculated. In the second method the response of the hull girder, that is its motions and/or strains, are measured and then, if it is desired, the loading which caused this response is inferred from the response. Let us examine the advantages and disadvantages of these two approaches.

A. *The Direct Method*

The load is of course the cause of the strains and motions of the ship. We can calculate the strains and motions if a valid theory of behavior under such loads is available. In order to compute this response it is necessary to know the load as a function of space and time; that is we must measure the pressures at a sufficient number of points on the ship so as to provide a reasonable picture of its distribution at any instant. This requires a large number of pressure gages with the attendant recording equipment, a costly, complex installation which is likely to be a source of much trouble and is certainly not suitable for statistical studies. Analysis of such data is a tedious job at best. Furthermore, for such data to be of general use in calculating the behavior of ships, it is necessary to relate the water pressures to the wave configuration, and of course a valid theory for making the calculations must be available. The calculations themselves might be too complex and time-consuming to permit their use even if the measurements and their analyses were practicable for statistical studies.

It is believed that this type of approach does not lend itself to studies which must extend over long periods of time. The method is however quite useful in checking the validity of proposed theories of ship behavior.

B. *The Inverse Method of Measuring the Ship's Response to the Wave Forces*

The response of the ship girder can be defined by its rigid body motion if the ship is to be regarded as a rigid body. If vibrations of the several parts of the ship relative to each other are also of significance, we need to know the natural modes of vibration of the ship, which may be obtained by means of a separate vibration generator survey in still water, and in addition we must know the motion of the hull relative to its rigid body position. It is probable that in the great majority of cases the significant stresses are obtained with substantial accuracy by assuming the vessel to act as a rigid body. The magnitudes and distribution of strains induced in the ship are a direct indication of the loading. Measurements of strains under actual service loads will indicate how well the vessel has been designed

to withstand the applied loads and will show the points of maximum bending moments on the basis of simple beam theory. The bending moment diagrams can be obtained directly from a knowledge of the stresses and section moduli. It can be shown that a knowledge of the linear acceleration of the center of gravity of the ship will permit one to approximate the wave shape passing the vessel. If at the same time the angular position of the vessel relative to the wave is known, the ship's position on the wave can be approximated and consequently the resultant stresses can be calculated. The number of measurements required to define the predominant motion of the ship are seen to be very few, for example, the pitching and rolling angles and accelerations as well as the vertical acceleration of the center of gravity together with the ship's heading relative to the waves. It is possible to construct apparatus which will obtain these data automatically over extended periods of time at relatively low cost. The type and number of measurements are such as to lend themselves to economical simple installations on many ships. The data have immediate significance in terms of load, stress or seaworthiness, and lend themselves to statistical analysis.

In addition to strain and acceleration measurements on ships, we may profitably measure directly the wave configuration occurring in various bodies of water. The measurement should be carried out over long periods of time in order to determine the possibility and probability of occurrence of given waves.

We will, in the remainder of this paper, deal with the second (inverse) approach which requires the measurement and analysis of ship response to unknown forces paralleled by collection of statistical wave data.

It was mentioned earlier that the complexity of dynamic ship calculations has retarded their use. This disadvantage has just lately been overcome by means of an electrical analogy to the structural problem. In this analogy the ship is divided into a number of stations. The mechanical properties of the vessel are calculated for each station and converted into corresponding electrical quantities. The force acting at each station on the ship as a function of time is applied at each station in terms of a corresponding voltage. The resulting shear and bending moment at any station can be measured directly as a function of time. The process is automatic and can be repeated rapidly for various applied forces (waves) or for different structural designs. The analogy includes the effect of vibratory response of the ship to the applied forces. It is believed possible also to determine by an inverse procedure the forcing functions (wave forces) which induce the measured bending moments.

We now come to the instrumentation required to obtain the necessary data. It will be advantageous to classify this instrumentation into two categories, namely:

(a) Instrumentation that provides data on a time basis, such as acceleration-time data.

(b) Instruments that provide a statistical analysis of the measured quantity. This analysis may be a very simple one, such as the total number of times that a given stress has been exceeded, or a more complex one, such as the total number of times that given variations of strain occurred, together with the average stress about which the variation occurred.

It should be our purpose to develop these instruments in such a manner that they will be as nearly automatic as possible and yet remain simple in construction and operation. It should be possible to install a set of instruments on a ship and then remove the data after an interval of say two months without furnishing operators or special care during the interval. The data should be in a form which requires a minimum of analysis for a usable evaluation. A brief description of a number of such instruments is given in the appendix.

RECOMMENDED PROGRAM OF INVESTIGATION

Let us now proceed to integrate the necessary measurements into a simple unified program for the collection and analysis of data. This program is to consist of four parts; namely, (1) preliminary sea trials, (2) statistical surveys, (3) systematic study of sea conditions, (4) statistical analysis of data. First there will be selected a very limited number of ship types for preliminary tests. Each one of these ships is to be instrumented to a greater extent than will be required for the subsequent mass surveys that are to follow. The main questions to be looked into during the preliminary tests are:

(a) At which longitudinal position on the hull do the maximum bending stresses occur?

(b) Is it necessary to investigate flexure of the hull in a horizontal plane as well as flexure in the vertical plane?

(c) Are fatigue stresses a major consideration?

(d) What is the probability of sustaining high impact loads on the bow?

(e) Does the vibratory response of a ship play an important role in the strength of the hull girder?

In addition to these special questions, we will require the following information which will be desired also for the subsequent statistical surveys:

(i) The rigid body motion of the ship which will permit an approximation of the wave profile passing the ship, the wave velocity and the position of the ship relative to the wave.

(ii) The history of the longitudinal bending stress at the midship section; the strain measurements are to be taken on the keel.

The third series of tests will consist of an extension of a systematic study of sea conditions based on data collected from instruments such as the DVL type instrument previously discussed, as well as from the data obtained from the ship studies.

When sufficient data have been obtained to permit definite conclusions as to the wave configurations and wave velocities to be expected in the various oceans, as well as to the questions stated above, we will be in a position to predict the strength of any normal ship girder subjected to these conditions either by using the method of the electrical analogy* which has been outlined earlier in this paper, or by comparison with the statistical criteria which will have been established for the particular type of ship and service under consideration. It is quite probable that the electrical analogy will be the quickest and most economical method of solution. As soon as the physical constants of the ship are available, solutions of the response to selected wave conditions should be obtainable in a few hours.

The instrumentation† to be installed for the preliminary sea trials would be as follows:

(a) Motion recorder.

(b) Strain cycle counters (four required) amidships to determine the bending stresses in the vertical and horizontal planes.

(c) Strain cycle counters (four required) to measure bending strains in the keel located at the forward ⅛, ¼ and ⅓ points and at the after ¼ point.

(d) Automatic strain analyzers (two required) one each to analyze the bending strains in the keel amidships and at the forward quarter point.

(e) A few pressure gages located at the ship bow and about three strain gages at selected locations. The output of these gages is to be recorded as a function of time. This particular apparatus is to respond to frequencies up to about 15 CPS. The purpose of these gages is to furnish information as to

*Beam Vibration Analysis With the Electric Analog Computer, by G. D. McCann, and R. H. MacNeal, Journal of Applied Mechanics, paper No. 49-SA-3.

† See Appendix.

the severity of impact forces acting on the bow, as well as to indicate the relative importance of vibratory stresses associated with hull vibration. The instrumentation to be installed for the collection of data on the statistical surveys* is as follows:

(a) One motion recorder.

(b) One strain cycle counter or one automatic strain analyzer; the choice will depend on the result of the preliminary trials.

It would not be too far-fetched to hope that this latter compact instrument installation can be accomplished on almost every major vessel afloat. The cost would be very small. The very fact that the shipowners are keeping themselves continually informed on the strains and seaworthiness of their ships, should add to their prestige. It is estimated that if the instruments here discussed were manufactured in quantity they would cost about $4000 for the motion recorder, $1300 for the strain cycle counter and $800 for the strain analyzer. It should be remembered that these figures are estimates and are given here for the purpose of providing a rough idea of the costs involved.

The data obtained from the motion recorder could be presented statistically by

(a) Relating wave periods to wave heights.

(b) Plotting the incidence of various wave heights as a function of the season and of the geographic location.

(c) Plotting maximum pitching acceleration and pitching angle as a function of wave length/height ratio and relative wave velocity. Many similar criteria can be investigated.

The data obtained from the strain cycle counter can be related to specific sea conditions or it may be more simply plotted as strain amplitude against frequency of occurrence for several mean strain levels. It is suspected that the mean strain level will be the still water strain.

The data obtained from the strain analyzer can be very simply presented as a graph of strain against the percent of the total operating time that the strain has occurred, or if preferred strain may also be plotted as the percent of the total operating time that a given strain has been exceeded.

Equipment for the study of wave conditions should be vigorously developed to the end that data may be obtained over long periods of time, say several months without requiring servicing of the buoys. Perhaps the most promising approach would be the transmittal of information from the buoy in the form of radio signals. These buoys could be built in conjunction with automatic weather stations distributed over the various oceans which would send out their reports automatically as radio signals.

It has been the intention in this paper to take stock of our present position with regard to the problem of the longitudinal strength of ships at sea, to examine existing material as to its usefulness for our purpose and then to arrive at a program for the study of the strength requirements of ships at sea, so as to permit them to withstand the forces acting on them under the service conditions that obtain. A definite program for the collection of test data has been presented which is believed to be economical of time and cost and which lends itself to the statistical study of waves and of the strength of ships at sea, with a minimum requirement of technical help. Existing instruments have been examined and where necessary—new instruments are now being developed in order to provide the more salient data, always remembering the objective of producing a simple reliable automatic instrument which will be easy to install, which does not require personnel for its operation, which does need

*This instrumentation alone would provide valuable data if it were felt necessary to reduce the scope of the studies.

but a minimum of technical help for data analysis and which is low in cost. This ideal instrument may not always be possible to develop, but it can be approached if these requirements are continually kept in mind. The cost of such a program should be borne, it is believed, jointly by the classification societies, the shipowners, the U. S. Navy and the several governments interested in the progress of commercial and naval shipping.

If I have been able to impress on the reader the desirability of the statistical study of sea conditions and of the strength requirements of ships at sea, I shall feel rewarded for having attempted this presentation.

APPENDIX

I. INSTRUMENTATION THAT PROVIDES DATA ON A TIME BASIS

Motion Recorder: This instrument is to record, on a single chart, the output of two accelerometers and two position indicators. The quantities to be recorded are: angular pitching acceleration, linear vertical acceleration at the ship's center of gravity, and the rolling and pitching angles. The equipment will operate automatically and the installation itself will be a simple one. The recorded data may be removed at desired intervals for analysis. On the basis of these data, it will be possible to plot the approximate wave shape and to fix the pitching position of the ship relative to the wave at any instant; the dynamic forces due to pitching are also readily computed.

Strain Gage: A strain gage now in the development stage incorporates the principle of a differential transformer. This gage is relatively insentitive to change in humidity and temperature and can be immersed in water or oil. The output of the gage will be sufficient to operate sensitive galvanometers directly without amplification, or it can be used in conjunction with a recorder of the type described under the heading of "Motion Recorder."

Pressure Gage:* This gage is to incorporate the same features as the strain gage preceding. It should be designed so that it can be installed in the skin of the ship so as to measure the water pressures acting at various points on the hull without requiring difficult installation procedures.

Wave Measuring Apparatus†: It will be advantageous for the statistical study of waves to have an apparatus that records wave periods together with the corresponding wave heights over extended periods of time. The DVL Wave Measuring apparatus is one of several suitable developments in this direction. It has been adequately described by Pabst in DVL report No. 360. This instrument takes advantage of the fact that the pressure variations due to surface waves are rapidly attenuated with increasing depth. Essentially the apparatus consists of a suitably restrained buoy which moves up and down with the waves. A pressure sensitive

*This gage is to be used for special studies of bow forces.

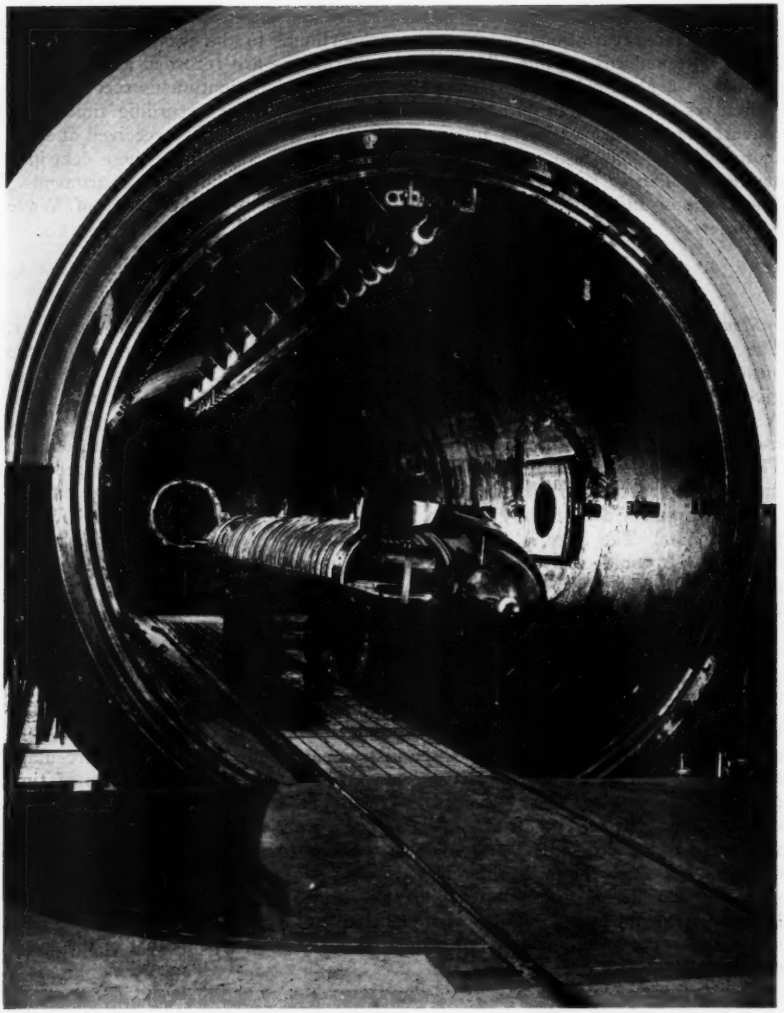
† This apparatus, although not measuring the response of a ship, is suggested here for special studies such as are conducted by the Scripps Institute of Oceanography.

device and a recorder are attached to the buoy by means of a long cable. The wave raises and lowers the buoy which in turn raises and lowers the pressure sensitive capsule causing variations of pressures proportional to the wave heights to be impressed upon this capsule. The pressure variations are recorded as a function of time on a glass cylinder. The present device provides continuous recording for eight days. It should be practicable to provide for longer recording time or to convert the pressure variation into a radio signal which may be received at some shore station. It is also possible to provide automatic analysis of these data in a manner similar to that which will be discussed under "Statistical Instruments." The Scripps Institute of Oceanography has developed several types of Wave Recorders.

II. STATISTICAL INSTRUMENTS

Strain Cycle Counter: This instrument consists of a strain gage and a counter. The counter tabulates the number of cyclic variations of strain together with the mean strain about which these cyclic variations occur. In addition the gage also gives the total number of cyclic variations independent of the mean strain level. A ten-inch strain gage is used in the present model. This type of gage will be useful in determining the strain history from the standpoint of fatigue loading. The readings of the counters could be photographed automatically at desired time intervals in order to correlate the strain variations with sea conditions and loading. The principles of this strain gage can be applied to the statistical analysis of other types of data. The gage is self-contained, easily installed and fully automatic in operation. It could easily be installed on a large number of ships to give a direct criteria of their loading. If the position of the neutral axis is known, a stress can be immediately converted to a bending moment.

Automatic Analyzer of Strains, Pressures, Accelerations, etc.: This apparatus is to determine (a) the total time that a quantity such as a predetermined strain has been exceeded and (b) the number of times that this strain has been greater than "A" and less than "A_1" for a continuous duration of time greater than "T" and less than "T_1". In other words, the instrument would give the total percentage of the time that given stresses have occurred as well as the percentage of time that a given stress has been exceeded. For example, it will probably be the general case in ships that a very large stress will occur only rarely and that low stresses will occur quite frequently. This analysis, if taken over a sufficiently long period of time, will show the probability of exceeding a given stress. This type of data could be obtained on different types of ships and over various sea routes at very low cost. The apparatus would consist principally of a number of time delay devices and a strain gage. The installation would be of the same type as that of the strain cycle counter.



Official Department of Defense Photo

A new installation known as the Ordnance Environmental Laboratory has been dedicated at the White Oak, Maryland, Naval Ordnance Laboratory. Depicted here is a deep sea pressure vessel which simulates one-half mile depth at sea. 15,000 gallons of water are pumped in the vessel during testing of under-water ordnance. A torpedo with cutaway side is shown prior to a test run.

PREVENTIVE MAINTENANCE THROUGH LUBRICATION CONTROL ON DIESEL ENGINES

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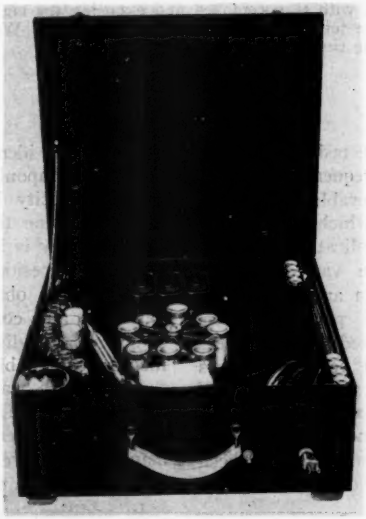
THE AUTHOR

Prior to and after his retirement from the U. S. Coast Guard, had a vital interest in used or operating oil and its relation to maintenance. His first contact with diesel propulsion came about through an assignment to the Winton Engine Company (now General Motors, Cleveland Diesel Division) in 1926. From that date all of his active duty assignment had to do with diesel maintenance and operation. Since retirement he has continued research work in the same field.



The Gerin Oil Inspection Kit, complete with tray containing all the necessary equipment and chemicals for testing lubricating oil contaminations.

The Oil Inspection Kit with tray removed and exposing the eight-place centrifuge head and 50 ml centrifuge tubes. Four each, Precipitation and Water and Sediment tests may be made at one time.

Confronted with the task of selecting suitable lubricants, frequently those in charge are in considerable doubt concerning the factors which need to be considered. In this application the diesel engine presents more varied lubrication requirements than any other class engine. The various problems have, however, been attacked by engineers and scientists over a period of years with the result that greatly improved products are now available and many of the most troublesome difficulties confronting the diesel operator have been solved through new oil specifications and characteristics.

All engineers are familiar with the tests developed for measuring the quality of lubricating oil which give some indication as to how the oil will perform in practice. Several of these tests are primarily for identification, i.e., appearance, odor, saponification, and ash. Viscosity, viscosity index and pour point tests determine the fluidity of the oil. The volatility is indicated by flash, fire, and carbon residue tests. Performance tests may be obtained which measure oxidation and corrosion-resistance, detergency and dispersion. Other tests include foam-stability, extreme pressure, reaction with water, homogeneity, and compatability. The above tests along with engine performance tests, and many others are used for predicting the performance of new oil.

In addition, operating engineers are provided with recommendations by engine builders, equipment manufacturers, and various governmental departments (e.g., Bureau of Ships Manual) concerning the specifications and charac-

teristics of new oil for use in their equipment. The oil supplier is continually working upon his product to supply the proper lubricant for any engineering problem.

However, regardless of how satisfactorily an oil performs in engine tests which closely simulate actual field operating conditions, the true test of a lubricant is its actual performance in every-day service. A test in one particular engine does not necessarily insure its satisfactory performance in all makes of engines. Before an oil can be definitely classified a suitable product, it should be subjected to extensive field tests in a wide variety of engines operating in those types of service for which the oil is intended.

From the first day in service every engine begins to develop its own lubricant diet. Frequently it is a matter of attention and care given by the operator. Often it is caused by geographic or climatic conditions. Certainly no two engines ever built have performed identically, worn out simultaneously, or required the same amount and kind of lubricant for the best performance.

The foregoing is given only to emphasize the fact that all fuel and lubrication engineers and oil technologists are continuously working on the problem of furnishing the proper lubricant for the particular installation and its application.

In discussing the conditions that are met by the lubricating oil in the cylinders, bearings, and the crankcase, the engine operator may be justifiably left with the thought that the lubrication situation is a very complex one and that deterioration of his engine, due to action and reaction in the lubricating system, may be occurring without his knowledge.

One interested in lubrication or efficient engine operation should devote a great deal of attention not only to the properties and characteristics of the oil recommended for his engine and supplied by the vender but also to what happens after a few minutes or hours of actual service.

Engineers and maintenance personnel are interested in two things with respect to the operating oil being used in their equipment: (1) Should the oil be continued in service operation and (2) What can the condition of the oil tell him about the functioning of his engine.

Establishing the frequency of oil change has always been difficult and each case should be considered individually. The rate at which the engine oil accumulates contamination from the atmosphere, operational conditions, or due to engine malfunctioning can be the principal criterion for the frequency of these oil changes. Various directives recommend limits for governing oil drain periods and such condemning limits are determined by frequent oil analyses and inspection of engine conditions. Where no analytical data is available, the drain periods are governed by operational hours.

Every experienced engineer is familiar with the engine indicator which measures pressures within the combustion chamber. From the cards taken or the pressures registered by these instruments a rather complete knowledge of what is happening inside the engine is obtained. Pyrometers reading exhaust temperatures provide a quick way to check the general condition of the engine. So long as the exhaust temperatures stay normal, it may be assumed that the engine is operating normally. Few are aware of another indicator from which conditions inside the engine may be determined—the analysis of the oil from the crankcase or circulating system. One of the chief values of this analysis is the measurement of contaminants, including foreign minerals, oxidation products, carbonaceous materials, dilution by fuel, and water.

It has been accepted that oil does not wear out but merely becomes contaminated which renders it unfit for further use. The damage caused by these contaminants is obvious, as they are all abrasive, with the exception of water.

The damage resulting from water is less obvious, but just as real. It is well known that water will rust iron and steel very rapidly but few people realize that this rusting can take place in the presence of oil. Oil containing a small percentage of water will not protect these metal surfaces when the oil is agitated. Further, the products of combustion, seeping past the rings, dissolve in any water present to form corrosive solutions. If other forementioned impurities are present, the water may combine with them and form sludge which may become so thick as to interfere with proper lubrication. Therefore, it can be readily seen that water can cause damage as serious as that resulting from other visible impurities.

Other impurities which are invisible in the oil are even more dangerous, as they complete their sabotage before they are suspected. These are the dissolved impurities. Just as it is impossible to distinguish between salt and fresh water merely by looking at them, it is likewise impossible to detect dissolved impurities by visual inspection. These impurities are acids, resins, or gums, and asphaltenes which are soluble to a certain extent. They will not remain dissolved for as they come in contact with the hot walls of the cylinders, further oxidation takes place causing them to become insoluble in the oil.

When this happens, there occurs a phenomenon known as lacquering, i.e., the gums and resins are deposited on the metal parts in the form of brown or black film which builds up, more or less rapidly, depending upon the amount of resin present in the oil. When this lacquer is first deposited, it is sticky and can be redissolved with fresh, highly solvent oil; thus, it can be temporarily removed by changing the oil, but the life of the new oil will be shortened. Unless the oil is changed soon after depositing, the resin changes from its sticky state into a hard insoluble glaze.

In its sticky state, the resin retards the action of the rings resulting in more blow-by and increased piston temperatures which aggravate the condition. When the resin sets to its insoluble form, the rings become stuck in their grooves and oil consumption, blow-by and wear increase rapidly.

Acidity is essentially free in practically all new oils. However, during use. acidity may develop as the result of high sulphur content in the fuel or oxidation of the oil. When a fuel containing sulphur is burned in the engine, a small percentage of the oxides of sulphur formed in the combustion chamber seep past the rings into the system of lubrication. If water is present in the oil, the oxides of sulphur combine with the water and form corrosive acids. This particular problem with today's fuel and maintenance ordinarily gives no trouble and very little increase in acidity due to sulphurous or sulphuric acid is manifested.

On the other hand, the acidity of all oils increase during service as a result of oxidation of the oil. The acids, being soluble in the oil, are insidious in their attack, since the only way to tell they are present in dangerous amounts is by a chemical test, or by serious corrosion that takes place in the engine. This has caused most operating engineers to take a passive attitude on the subject and usually acidity has received attention only through laboratory reports and conditions of relatively severe corrosion. The tendency to oxidation is normal when oil is in contact with air, a tendency which is greatly accelerated in the diesel engine because of large amounts of oxygen and heat. This causes deposits that will vary widely with different oils and temperatures, some appearing as hard carbons, some as soft, flaky carbon and others as sticky varnish.

When these decomposition products of the oil (resins, gums, acids, etc.) combine with contaminants (dirt, carbon, etc.) they form what is commonly known as sludge.

This gives a brief picture of contaminants that may be found in operating or used oil.

Oil venders and engine builders realize the need of used oil analysis by which physical, chemical, and mechanical tests indicate and measure the changes that have taken place in their particular product due to operating conditions. From these tests their engineers can, by study of these changes, estimate what the conditions are inside the engine.

Analytical data and its compilation concerning the condition of operating oil is generally accomplished by sending samples of the oil to an outside laboratory or yard and the result from the analysis, in the form of a report is returned to the user. These reports do not tie in directly with preventive maintenance through lubrication control for such outside analyses are usually too infrequent; the answers too slow in coming; and the significance of isolated analyses are partly or completely obscured by time lag.

Many papers have been presented by lubrication engineers on the significance of used oil analysis. Comments on such papers have invariably asked what the indications were in a used oil analysis that would show the operator what might be going wrong in the engine itself which requires correction in order to insure that the quality of lubricating oil is being maintained at a desirable level. Further comments have been noted regarding the tendency of engineers to search for a different grade of lubricant and a means of correcting a difficulty that frequently develops to be a mechanical discrepancy; which discrepancy had to be first corrected before any grade of lubricant could have functioned.

Diesel engines have two distinct lubrication requirements—the bearings and the cylinders, and each lubrication requirement demands specific qualities in the oil used. In addition, the lubricating oil must serve to carry away part of the heat developed in the engine, particularly if it is used for piston cooling.

There is no particular problem in the lubrication of the crankshaft and rod bearings in a diesel engine. Although large piston thrusts are transmitted by the piston rods and connecting rods, the bearing areas are sufficiently ample so that unit pressures are not excessive. The oil, being pressure fed to the bearings in large volumes, builds up a full lubricating film in the bearing clearances through rotation of the crankshaft. Unless mechanical faults in the bearing disrupt this film, lubrication is continuous and the film of oil separates the mating bearing surfaces to prevent wear.

Somewhat different conditions prevail at top end of the connecting rod where it is attached to the wrist or piston pin. As the connecting rod oscillates through only a small arc on the circumference of the piston pin, full rotation is impossible, and absence of the oil pumping action that accompanies a steady rotation makes the buildup of an oil film more difficult at this point. A full supply of oil under adequate pressure is therefore necessary if wear on the piston pins and bushings is to be avoided. This problem of effective lubrication is further aggravated by poorly maintained mechanical conditions, excessive temperatures, or contaminations.

Since the life and performance of crankcase oil is limited by oxidation and contamination, the simple problem of lubricating the bearings becomes more complex when the same oil is used to lubricate the cylinder walls. There is always a tendency toward a small amount of oxidation in the crankcase as the oil contacts warm air, or localized hot spots but this oxidation and heat destruction is much more rapid when the oil is subjected to the severe conditions on the cylinder walls. There is also an increased amount of contamination from the blow-by of combustion products.

The oil used in cylinders must serve

the purpose of lubricating the walls to prevent excessive wear and contribute toward sealing the piston against gas pressure loss. With each power stroke some of the oil on the cylinder wall is subjected to the extremely high temperature of the combustion in the cylinder. A proper oil for such duty should be one that will burn away the least during its brief exposure and, in so burning, form the smallest amount of carbon. The unburned portion must maintain sufficient body to provide lubrication until the supply is replenished.

When the injection timing is incorrect in some engines, all the fuel may not be burned, and it may find its way past the rings into the crankcase. A similar effect is achieved when faulty injectors permit dribbling of the fuel into the cylinder. On starting, this unburned fuel, in liquid or vapor state, may wash part of the lubricating oil from the cylinder walls.

Compression rings stick due to formation of carbon deposits in the ring grooves and corresponding ring surfaces. Some of these deposits may come from oxidation of the oil and others from the carbon and resinous products due from incomplete combustion. An accumulation of deposits in the ring grooves causes sluggish operation and eventual sticking of rings. When this happens the rings no longer make the proper seal with the cylinder wall and the combustion gases blow by. Piston rings sometimes break when the deposits are heavy or so uneven that the rings do not seat uniformly in the grooves.

On the compression stroke of a diesel engine a limited amount of air is forced past the top compression rings and contacts the oil on the cylinder walls and pistons. Since this air is at high pressure and temperature, it promotes some oxidation. As there is not sufficient air available to support complete oxidation, it is agreed that only a certain amount of high temperature oil breakdown occurs. However, as a result of this effect a thickening in the oil occurs and complex organic products are developed which form a resinous or carbonaceous film on the piston skirt known as varnish.

All diesel engine pistons will gradually accumulate some skirt varnish which is not particularly harmful unless it becomes too thick. When too thick it is scraped off by action of the piston and has a tendency to stick the oil rings and plug the oil control holes in the rings. This is followed by increased oil consumption because the oil wiping ring action fails and too much oil is left on the cylinder walls where it is carbonized.

There is no question but what the condition of the oil and its contents are indicative of many troubles and by testing samples removed from the system, remedial measures may be taken when the results of the tests are revealed. When all the items are fully correlated the results of periodic tests is an excellent basis for preventive maintenance, achieved at a minimum expense. When an oil is changed without first remedying the fault of contamination that is responsible for the poor condition of the oil, the same results can be expected to recur immediately after the change.

A recent paper on diesel maintenance notes in part that suitable analysis of lubricating oil can indicate to a remarkable extent what the general condition of the engine may be, thus enabling the operator to extend the period between progressive maintenance and overhauls. From the time an engine is placed in operation a suitable record should be kept to detect any abnormal changes in the characteristics of the engine, particularly in the lubricating oil. Such records are generally more reliable and useful than information now being recorded such as fuel consumption and exhaust temperatures since these latter characteristics are usually more difficult to interpret and samples are not available for rechecking. The

author of this particular paper further states that during analyses of lubricating oil by laboratories conventional checks such as impurities present, viscosity, flash point, etc., are not sufficient to correctly diagnose the condition of the engine. To obtain a reliable diagnosis certain conditions should be adhered to which might include the following:

1. Records must be systematic with periodic tests made at uniform intervals.

2. The oil specimen should be removed from the same point in the system and in each case under nearly identical conditions of operation. This insures the sample will contain sufficient amounts of such foreign matter as may be present in the engine. The important indication here is any sudden or progressive INCREASE in proportion of such matter, even though the actual percentage may be small.

3. The form for reporting the results of the analysis should show only the essentials and give recommendations on such items as frequency of oil and filter changes, and note such mechanical attention required and the action taken. Evidence of bearing metals and other impurities with all possible sources should also be given.

A careful analysis of the lubricating oil, properly interpreted and made at frequent intervals would show the extent of the following conditions:

(a) Fuel dilution, caused by leaky injection pumps, or improper function of injector nozzle.

(b) Soot from smoke, due to overload, improper fuel injection, or clogged filters.

(c) Water from leaks or condensation (low temperature of cooling water or wall temperatures).

(d) Wear of cylinder sleeves, piston, and rings because of sticking or improper lubrication.

(e) Bearing wear or overheating because of improper installation, lubrication, faulty material, or engine operation.

(f) Oil filters having clogged elements or improper type.

(g) Oxidation because of excessive temperatures, clogged coolers, or insufficient flow.

There are three broad purposes to be served by routine measurements of the contaminations in lubricating oil. They establish a periodic week to week, or month to month record as regards contaminants which may be suspected of contributing to operating troubles, wear, etc. They are a necessary guide to purification practices needed to maintain an acceptable level of cleanliness of lubricating oil. Finally and perhaps of greatest interest, a change or increase in contaminants gives advance information that conditions requiring mechanical or operating attention are setting in.

The four tests which may be considered of great importance to the efficient operation of the engine and their maintenance in the eyes of operating engineers are: Precipitation; Water and Sediment: Neutralization (acid) Number, and Viscosity Comparison. Any increase in these contaminants indicate changes are taking place inside the engine and the operator is thus informed that the engine is in need of attention.

A practical oil testing kit has recently been developed and marketed whereby on-the-spot analysis, within the skill of the operating personnel, may be made in the engineroom, always under the user's control, as it should be.[1] This equipment weighing approximately thirty-two pounds is self-contained and provides positive measurements of the four dangerous contaminants. Four samples of oil can be tested in 25 minutes or eight samples in 40 minutes and although no technical skill is required, results are in agreement with laboratory analyses for the principles employed are conventional in this practical test.

[1]Reprint of an article from "Motorship" by F. L. Gerin, Gerin Corporation; JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS, INC.: November 1948 issue, page 691.

Precipitation Number

This test (Fig. 1) indicates the amount of asphalt, resins, and gums which result from the inevitable chemical changes in the oil itself and from fuel blow-by. They have no lubricating values of their own, and they interfere with lubrication by being attracted to the metal surfaces in preference to the oil. The quantity is of prime importance since they are the chief factor in the formation of engine deposits and the binder material for particles of dirt, free carbon and metal.

The entire group of deposit forming substances is insoluble when the oil is mixed with special naphtha, or solvent. To make the separation the diluted sample is centrifuged for ten minutes. The centrifuge tube is graduated and the amount of insolubles contained in 10 ml of oil is called the Precipitation Number.

Water and Sediment Test

In this test (Fig. 2), the oil sample is mixed with a solvent which holds the gummy, asphaltic materials in solution but throws out the solids and water.

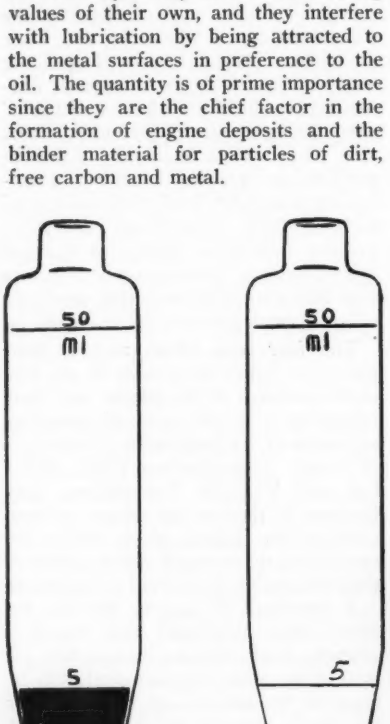


FIG. 1.—Test for Asphalt and Gummy Materials:

Put oil in glass tube to mark 5, naphtha to mark 50. Centrifuge ten minutes. Read amount of deposit from graduations on tube.

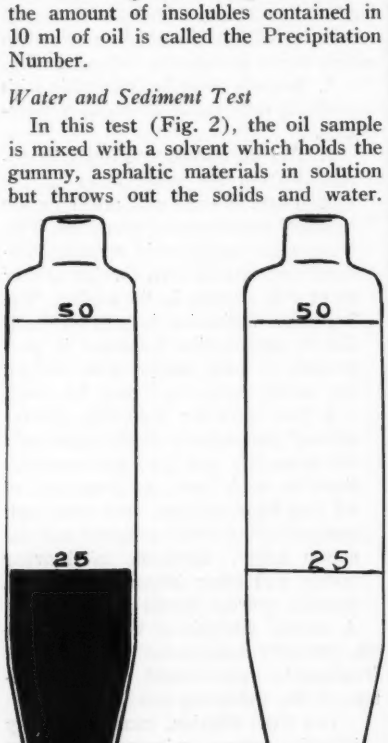


FIG. 2.—Test for Dirt, Sediment and Water:

Put oil in to mark 25, benzene to mark 50. Centrifuge 10 minutes. Read amounts of water and solids from graduations on tube. Note kind of solids and metal particles.

Whereas the precipitation or asphalt test measures self-made contaminants, this test measures dirt, soot from fuel, and water which might find their way through operation into the oil. Black sediment is assumed to be fuel soot or carbon. Metal particles, when present, can be seen at the very tip. Water, when present, causes a sharp sudden rise in deposits and a muddy color, or it may appear as a clear layer above the sediment. The water and sediment test is expressed in percent of oil.

Neutralization Number

It is unfortunate that acids are invisible in oil and their chemistry is somewhat complex. The Neutralization Number of an oil, an indication of its acidity, is the weight in milligrams of alkali—potassium hydroxide (KOH)—needed to neutralize the acid content of one gram of the oil. The general procedure for this test is to mix a 10 gram sample of oil with an alcohol solution—ethanol, to which 10 percent by volume of methanol has been added—and distilled water equal in volume to the alcohol solution. To this mixture one milligram of phenolphthalein, a color indicator, is added. Small amounts of potassium hydroxide are then added under specified conditions until the pink color, resulting from this addition, persists. This indicates that the acid content of the oil has been completely neutralized.

It is obvious that this method is not practical for engineroom use and the practical kit contains ampules in strengths of equivalent neutralization numbers of .3, .6, and 1.5 (mg of KOH per gram of oil), and a prepared acidity solution. An oil sample is placed in a glass cylinder (Fig. 3), to desired mark and the prepared acidity solution to 50 ml mark. An ampule of desired strength KOH is then broken and poured into the mixture. After shaking the mixture, if the solution remains pink the sample passes. If white or yellowish the sample fails. If one had the occasion to measure

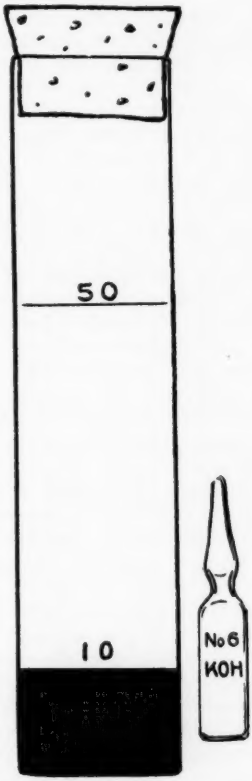


FIG. 3.—Acidity Test:
Put oil in to mark 10, test solution to mark 50, and one ampule of alkali. After shaking, if solution shows pink, oil sample passes. If not pink, sample fails. Ampules equivalent to neutralization number of .3 and .6 furnished for power plants.

acidity at levels other than the ampule ratings, a range from .2 to 3.0 mg may be obtained by varying the amount of oil sample.

There is no definite maximum operating limit on neutralization number for all used oils. Much better utilization of this test can be made by keeping a record of successive determinations. Over long periods each sample should show a neutralization number close to

the preceding one. When serious oxidation takes place, the values will increase rapidly. If there has been no dilution, the increase in neutralization number is generally accompanied by an increase in viscosity. It frequently happens, however, that the thickening of oil from oxidation is counteracted by sufficient fuel dilution to hold the viscosity to that of a new lubricant or lower. A high neutralization number indicates that oxidation of the lubricating oil is occurring which may be accompanied by the formation of varnish deposits and sludge. Low temperatures sludge formation from oxidized oil, which is accelerated by presence of water, may plug oil lines and damage journal bearings.

The passing Neutralization Number, or increase of acidity since last test, may be found by putting additional ampules of KOH in the mixture, recording each failure and the value finally passed. When the asphalt (precipitation number) is high the acidity can also be expected to be high. Whenever the oil shows an appreciable amount of asphalt from the centrifuge tests, the acidity test should be delayed until the oil has been clarified by filter change or centrifuge.

The additives in modern oil act upon the KOH for measuring acidity and thereby produce a relatively high initial reading for the new oil, which is not acid in true sense. A practical test for engineers should then be used to determine what is normal for the new oil and then test for any increase above normal.

Viscosity and Fuel Dilution

Contrary to belief, oils, of themselves do not thin out in service but rather tend to thicken. However, in service most oils show a lower viscosity after use than in new condition due to fuel dilution.

The most commonly used instrument for determining viscosity is the Saybolt type with universal orifice. Viscosity, expressed in seconds (Saybolt Universal) is the time required for 60 milliliters of the liquid to flow through the standard aperture of the instrument at a stated temperature. The viscosity of

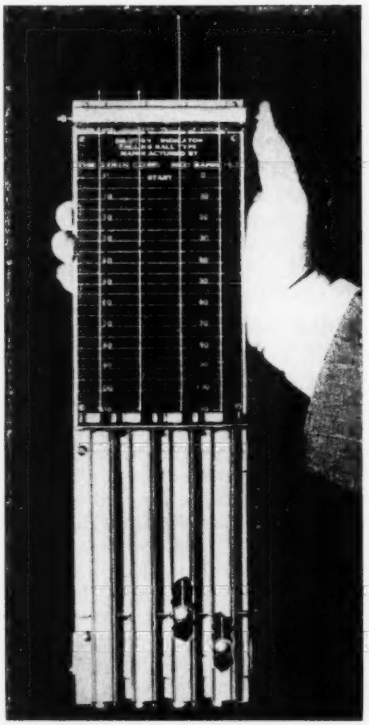


FIG. 4.—Viscosity and Fuel Dilution Indicator.

Test for Fuel Dilution and Viscosity: Change in viscosity is used to show extent of fuel dilution. The dilution indicator has four balls falling in tubes filled with the oils. The balls can be dropped in pairs, threes or fours. The balls are started from "0" by manual release button and stopped in flight when one ball reaches "100". Assume, for instance, the ball in a used oil stopped at "100", and the ball in the original new oil at "60". This used oil has 60% of its original viscosity. Nine tubes are furnished. As many as eight samples of used oil can be tested in quick sequence.

oil changes with temperature and this rate of change may be ascertained by the Viscosity Index.

Fig. 4 shows a viscosity and fuel dilution indicator employing the falling ball principle for comparison on the new oil and a used oil of similar grade. One tube marked A contains new oil sample whereas the remaining tubes contain used or operating oil. A change in viscosity is shown by the fall of the ball in the used oil compared with the fall of the ball in the new oil. The balls are released at "0" by manual button and stopped in flight when one ball reaches "100." The reading reached by the slower ball applies to the oil with the fast ball. Temperature is the most critical factor in all viscosity tests. Oils that have reached room temperature before filling the metal tubes require approximately 15 minutes to equalize whereas samples just brought in require 30 minutes. This test is a comparative test and requires no index or interpolation. The results are expressed in percentage of used oil viscosity to that of the new oil, and results are accurate for maintenance purposes without undue concern over temperatures.

To determine actual fuel dilution instead of percentage original viscosity, test cases may be made by the engineer. Fuel is added to new oil in certain proportions, and the viscosity compared with the original oil. By comparing these mixtures of fuel diluted oil with undiluted new oil, each operator can establish what viscosity changes mean in terms of percent dilution for his particular oil. This is of interest when the condemning limit has been established.

It is not the intent of this paper to impart knowledge which will enable the operating engineer to do the work of a chemist or oil technologist. The object is to convey practical information useful in securing better results in diesel operation through lubrication control. Recording the results of on-the-spot analyses establishes a continuous record of proper lubrication, or shows trends

LUBRICATING OIL CONTROL RECORD

Vessel: U.S.S. MAINTAIN | Engine: EFFICIENT - EY | Serial No. 146739 | Capacity 400 Gals.

Date 1950	Oil Symbol No.	Oil Hours: Since Last Report	Oil Hours: Total	Filter Hours: Since Last Report	Filter Hours: Total	Viscosity % of New	Ppt. No.	Water & Sediment: Sed. %	Water %	Water & Sediment: Metal	Neut. Nos. Failed and Passed	Make of Oil: Mechanical Attention Indicated, Action Taken and Result (below)
6/1	9250	70	160	70	160	96%	T	N	N	N	P. 3	Slight dilution indicated. Check injectors
6/15	9250	120	280	120	280	94%	.05	LT	N	N	F. 3 P. 6	Fuel dilution with ppt. increase. Check combustion.
7/2	9250	150	430	150	430	94%	.1	.1	N	N	F. 3 P. 6 P. 9	Change filters & check injectors. # 2 injector changed. Check fuel dilution within 24 hours.
7/3	9250	20	450	20	20	92%	N	N	N	N	F. 3 P. 6	Fuel dilution decrease. Oil OK.
7/20	9250	209	659	209	229	96%	N	T	T	N	F. 3 P. 6	Trace of water by muddy color. Check system and temperatures .
7/23	9250	40	699	40	269	98%	N	N	N	N	F. 3 P. 6	No water in evidence - Oil OK.
8/23	9250	310	1009	310	579	90%	.05	.1	T	N	F. 3 F. 6 P. 9	Fuel dilution approximately 2%. Check for dilution in two days. Change filter.
8/25	9250	40	1149	40	40	92%	.2	.2	N	N	F. 3 F. 6 P. 8	Change oil. Check for oxidation through high temperature and blow-by.
9/1	9250	100	100	100	100	100%	N	N	N	N	P. 3	Overload and high temperatures were apparent.
						EXAMPLE EXAGGERATED						

T = Trace N = None LT = Large Trace

which can easily be interpreted. Making frequent tests at regular inspection periods quickly establishes what is normal for the various values and the interpretation is merely a question of noticing when the values depart from normal. As the centrifuge results increase to readable values, and irrespective of the size of deposit, it indicates filters are becoming "loaded." A progressive increase in fuel dilution indicates leaking or dribbling injectors.

It is often erroneously thought that the quality or performance of the lubricating oil causes or contributes to crankcase explosions. In most cases upon examination, the cause has been found to be a faulty mechanical condition caused by a hot spot or localized temperature in the engine which was sufficiently high to ignite a mixture of air and oil vapor. To guard against possible explosions fuel dilution should be held to a minimum.

Various other operating factors which are detected through their effects on the lubricating oil will occur to the reader. All in all, a practical on-the-spot analysis of lubricating oil offers a great opportunity to know what is going on inside the engine. Moreover, the ability to measure and read actual data on the lubricating oil has also a marked psychological effect upon operating engineering personnel, stimulates beneficial interest, and leads to more economical operation and maintenance.

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AN ANALYSIS OF TEMPERATURE DISTRIBUTION IN A METAL HOLLOW CYLINDER OF FINITE RADIUS UNDER UNSTEADY STATE CONDITIONS FROM E.E.S. DATA

LT. COMDR. ROBERT J. KNOX, U.S.N.

THE AUTHOR

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INTRODUCTION

The U. S. Naval Engineering Experiment Station, Annapolis, Maryland has run tests over a period of years to determine the effects of thermal shock on steam piping for naval use. The calculations of the stresses set up by thermal shock, however, require a knowledge of the temperature distribution in the wall at all radii. Just what the temperature distribution is in the pipe wall before steady state conditions are attained is the subject of this investigation.

Thermal shock in the operation of a Navy steam propulsion plant is an abnormal condition. The possibility of thermal shock conditions, through a casualty, is however always present. High temperature shock can be the result of failure to warm up a main steam line properly before admitting high pressure saturated steam from the boiler. Cold shock can be the result of "carry-over" of a water slug from the boiler into the lines which have been carrying superheated steam.

The author first became aware of this problem in August 1947 during an informal discussion with Mr. W. C. Stewart of the EES. By this time the EES had obtained a sizable amount of data from cyclic thermal shock tests. These tests were run on various sections of pipe to determine the physical

effects of cyclic thermal shocking. This data is used as a basis for an investigation into the subject problem.

The author wishes to express his deep appreciation to those who have aided in this investigation. He is especially grateful to Associate Professor George H. Lee, U. S. Naval Postgraduate School, for his assistance and continued guidance. The author is also grateful to the Engineering Experiment Station and its Mr. William C. Stewart for suggesting the investigation and cooperating to the fullest in furnishing data. He further expresses appreciation for the counsel of Doctor Warren M. Rohsenow, Massachusetts Institute of Technology, Cambridge, Massachusetts.

DATA

Tests were run on a length of 25-20 Cr-Ni, 3/4 inch IPS, schedule "double extra strong" pipe using saturated steam, superheated steam and exhaust gas as the heating media. The specially bladed turbo supercharger wheel at the EES was employed to supply the combustion gas using gas temperatures ranging from 1040F to 1360F. Other runs were made by passing superheated steam at 500 psi pressure and 1050-1070F temperature through the sample. A third set of test runs was made by passing saturated steam at 500 psi pressure and 470F temperature through the test pipe. The resultant curves of inside and outside temperatures vs. time for these runs are shown in Figs. 1, 2 and 3 respectively.

One thermocouple was peened to the outside wall at mid length of the tube. Another thermocouple was brazed to the bottom of a ¼ inch diameter hole drilled to within 0.01 inch of the inside wall, ⅛ inch distance from the peened-in thermocouple. In the test with exhaust gas, however, the latter thermocouple was pressed against the bottom

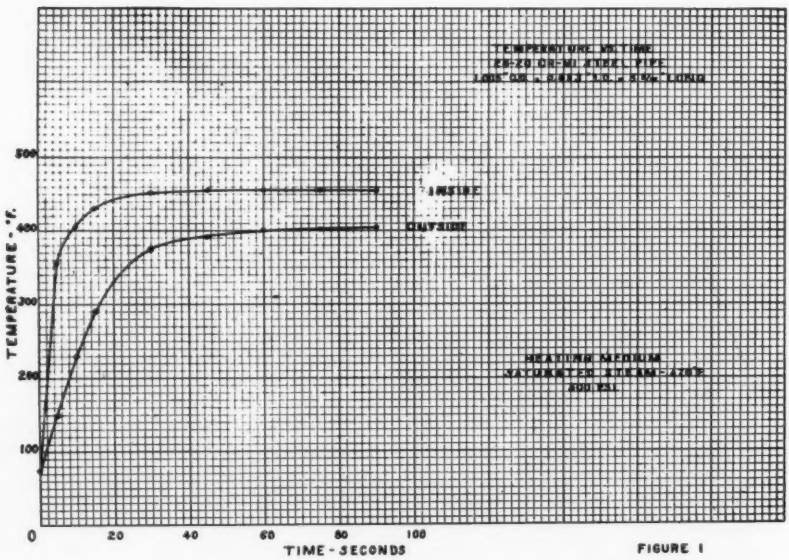


FIGURE 1

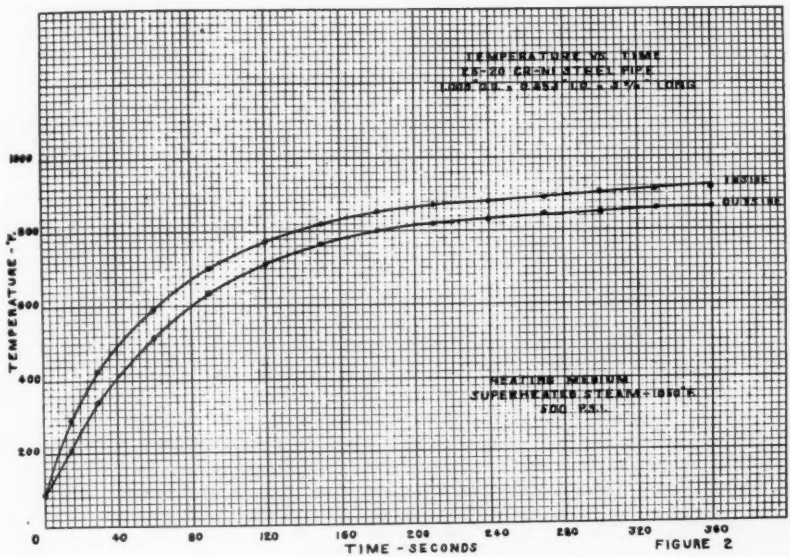


FIGURE 2

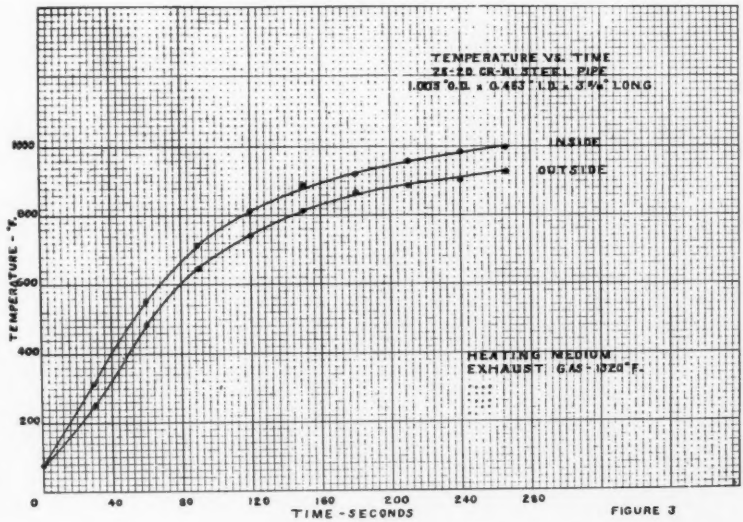


FIGURE 3

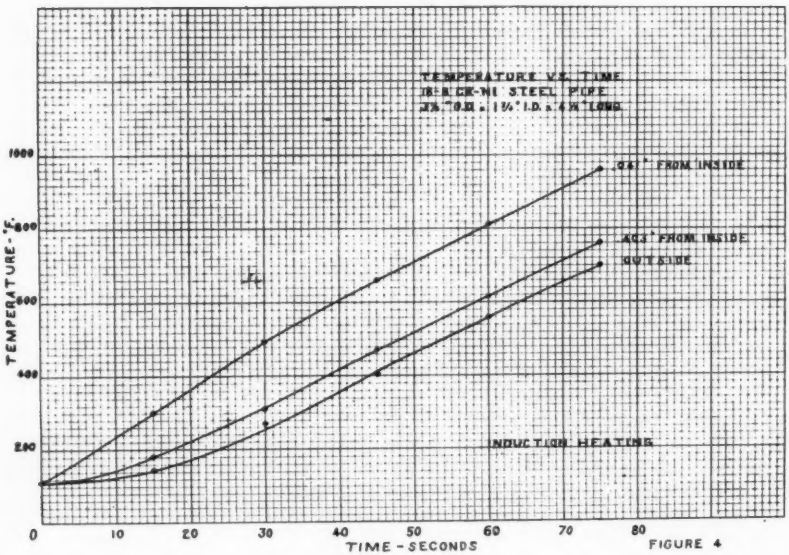


FIGURE 4

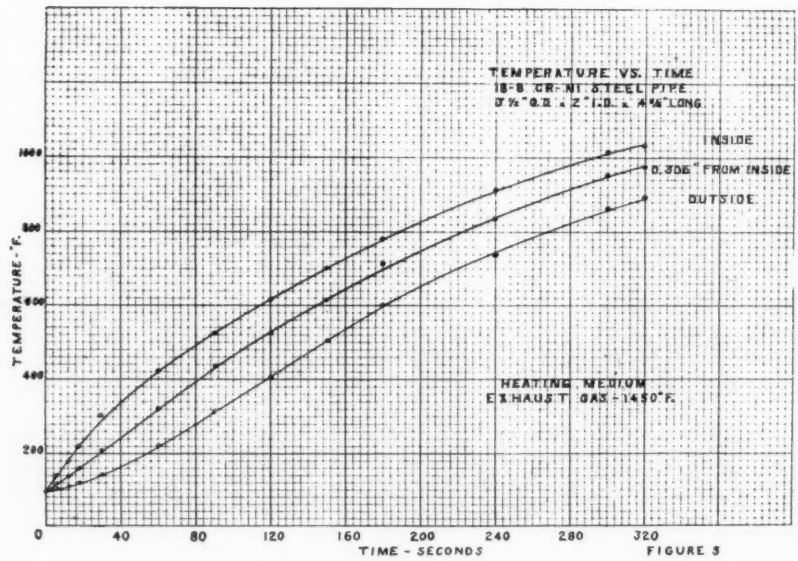


FIGURE 5

of the hole by means of an 18-8 Cr-Ni steel screw insulated from the thermocouple. The thermocouple wire used was No. 29 B & S gage iron-constantan. The inlet and outlet pipes carrying the heating medium were drawn up tightly against the counterbored ends of the test sample by means of bolts secured to flanges welded to the pipes. Insulating gaskets were placed in the counterbored ends of the test sample to minimize heat transfer due to conduction. Pipe lagging for insulation, however, was not used in any of the tests.

In addition to these runs a series of tests was made on a similar sample of 18-8 Cr-Ni steel pipe, 3¼ inch O.D. and 1¾ inch I.D. with thermocouples located .041 inches and .403 inches from the inside wall and a third peened to the outside wall. Thermal shock was created by means of induction heating. A plot of the temperature vs. time curves for all three thermocouples is shown in Fig. 4.

Runs were also made on an 18-8 Cr-Ni pipe, 3½ inches O.D. and 2 inches I.D. with thermocouples located .01 inches and .306 inches from the inner wall and a third peened to the outer wall. Thermal shock was created by means of exhaust gas at temperatures ranging from 1260-1450F. A plot of the temperature vs. time curves for all three thermocouples is shown in Fig. 5.

The results of these tests clearly indicated that saturated steam produced the greatest rate of temperature rise. Thermal shock tests were then planned using saturated steam as the heating medium. A section of two inch 25-20 Cr-Ni pipe 1.189 inches outside radius, .749 inches inside radius was set up for test in the same manner as the initial tests. Thermocouples were located as for the previous test runs with saturated steam. A representative plot of the resultant temperature curve is shown in Fig. 6.

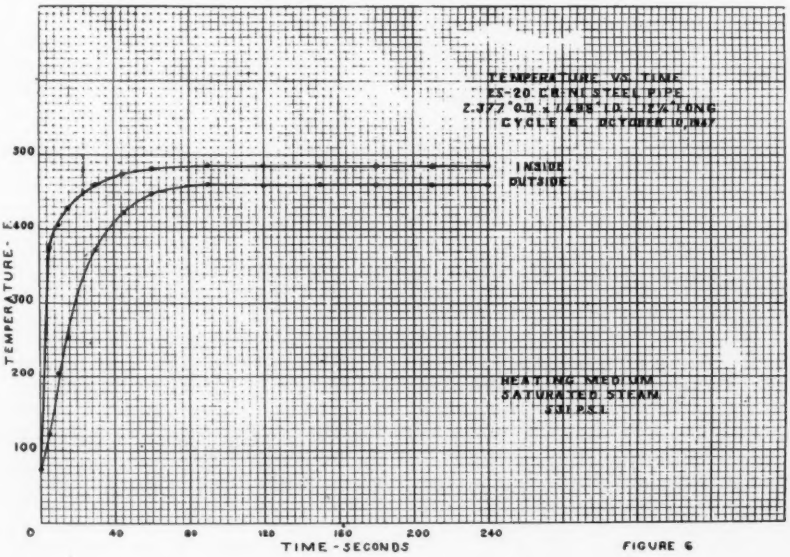


FIGURE 6

TABLE OF SYMBOLS AND ABBREVIATIONS

B—British thermal unit.
c—Specific heat—B/lb/F.
E—Modulus of elasticity in tension and compression.
J_n—Bessel function first kind, order n.
k—Thermal conductivity — B/hr/ft²/ft/F.
N_n—Bessel function (Neumann) second kind, order n.
n—Any integer.
r—Radius.
r_a—Inside radius of a hollow cylinder.
r_b—Outside radius of a hollow cylinder.
t—Time.
α—Diffusivity—$k/c\rho$ in ²/sec.
β—Coefficient of expansion.
Δr—Small finite difference in radius.
Δt—Small finite difference in time.
θ—Temperature.
θ_a—Temperature at inside radius.
θ_b—Temperature at outside radius.
$\theta_{r\pm\Delta r}$—Temperature at same time at $r+\Delta r$ and at $r-\Delta r$.
θ_t—Temperature at time t.
$\theta_{t\pm\Delta t}$—Temperature at time $t+\Delta t$ and $t-\Delta t$ at any one radius.
$\theta_{\lambda\pm\Delta\lambda}$—Temperature at same time at $\lambda+\Delta\lambda$ and $\lambda-\Delta\lambda$.
λ—ln r/r_a.
υ—Poisson's ratio (unit lateral contraction/unit axial elongation).
ρ—Density—lb/in³.
σr—Radial stress.
σt—Tangential stress.
ϕ—Any function.
B&S—Brown and Sharpe.
EES—Engineering Experiment Station.
I.D.—Inside diameter.
IPS—Inside pipe size.
ln—Logarithm to Naperian base.
O.D.—Outside diameter.
psi—lbs/in².
∞—Infinity.

ANALYTIC SOLUTION FOR TEMPERATURE DISTRIBUTION

To calculate the thermal stresses in a finite hollow cylinder the following formulae given by Timoshenko (14) are used:

$$(1)\quad \sigma t = \frac{\beta E}{(1-\upsilon)\, r^2}\left\{\frac{r^2+a^2}{b^2-a^2}\int_a^b \theta r dr + \int_a^r \theta r dr - \theta r^2\right\}$$

$$(2)\quad \sigma r = \frac{\beta E}{(1-\upsilon)\, r^2}\left\{\frac{r^2-a^2}{b^2-a^2}\int_a^b \theta r dr - \int_a^r \theta r dr\right\}$$

It is not the author's intention to show the derivation of the above formulae since they follow a straight forward

method and are presented clearly by Timoshenko. The only purpose in introducing them is merely to show their relation to the temperature distribution in a thick cylinder.

To solve the integral $\int \theta r dr$ it is necessary to obtain the temperature distribution through the pipe wall. It is therefore necessary to determine the analytic solution or to use a graphical method of solution.

Barker (1) and Lees (7) give the steady state temperature distribution in a hollow cylinder which is of interest but of little value. Churchill (3) and Williamson and Adams (15) present papers which give the analytical solution for the transient temperature distribution in a solid cylinder. The initial temperature is uniform. The surface temperature rises instantaneously to a given temperature and is maintained at that value. Carslaw and Jaeger (2) give a solution for the case of the infinite hollow cylinder.

Dahl (4) gives a solution for the case of the finite hollow cylinder. The initial temperature is uniform. The inner surface temperature is instantaneously raised to a value and maintained at that value. The outer wall temperature is maintained at the initial temperature.

Carslaw and Jaeger (2) also present the case of a finite hollow cylinder. The inside surface temperature is raised instantaneously and maintained constant. The outer wall is adjacent to an infinite fluid which maintains its temperature constant at a point distant from the outer wall.

Literature dealing with the analytical solution of temperature distribution in a hollow cylinder therefore is limited to highly idealized problems. Peculiarly, all published solutions have one common boundary condition, namely an instantaneous surface heat source. This is a condition in which the surface is instantaneously brought to a given temperature and then maintained at that temperature. This condition is not present in the actual problem.

The combination of three conditions is the cause for the complexity of this particular problem. These three conditions are: 1) steel pipe has a high thermal conductivity, 2) the heat source in the actual problem produces a variable surface temperature and, 3) the heat source produces a variable rate of change of the surface temperature.

In addition, the problem primarily is centered around the initial two or three second period of time. Boundary conditions used by Dahl (4) or Carslaw and Jaeger (2) cannot be assumed. On the other hand the problem cannot be dealt with as in the case of Perry and Berggren (10) who choose the cork lagging on a metal pipe and then work only with the cork cylinder.

Graphical solutions in the literature are developed for the elementary boundary conditions only. For example, the Schmidt graphical method for a slab (9) (11) (12) can be adapted to a cylinder wall but large errors are introduced during the early stages of the solution. The theory of these, however, can be applied to the actual problem. The author carries out this application for the case of heating only and presents results which he feels are accurate.

The maximum thermal stresses will occur in the early stages of heat transfer. It is therefore readily seen that an analytic solution is more desirable than a graphical solution.

1. The general solution.

Assume the initial uniform temperature of the body as zero, the boundary conditions of the problem are:

at $t = 0$ and $r = r_a$; $\theta = 0$
at $t = 0$ and $r > r_a$; $\theta = 0$
at $t = t$ and $r = r_a$; $\theta = \phi_a(t)$
at $t = t$ and $r > r_a$; $\theta = \phi_r(t)$

The differential equation for heat conduction in a cylinder (Fourier-Poisson) when radial changes only are observed is:

$$(3) \quad \frac{\partial \theta}{\partial t} = \alpha \left(\frac{\partial^2 \theta}{\partial r^2} + \frac{1}{r} \frac{\partial \theta}{\partial r} \right)$$

A general solution is obtained by assuming $\theta = R\,T$ where R is a function of r only and T is a function of t only. Differentiating and substituting:

$$(4)\quad \frac{1}{\alpha T}\frac{\partial T}{\partial t} = \frac{1}{R}\left(\frac{\partial^2 R}{\partial r^2} + \frac{1}{r}\frac{\partial R}{\partial r}\right) = -\mu^2$$

where μ^2 is any arbitrary constant.
A general solution therefore since $T = e^{(C_1 - \alpha\mu^2 t)}$ and $R = A_1 J_o(\mu r) + B_1 N_o(\mu r)$ is:

$$(5)\quad \theta = \sum_1^\infty \left[A J_o(\mu r) + B N_o(\mu r)\right] e^{-\alpha\mu^2 t}$$

where A and B are constants of integration and would normally be evaluated by substituting the boundary conditions. This procedure fails in this problem since all boundary conditions cannot be substituted to meet the final general equation after A and B have been evaluated.

Another particular solution is to let $\partial\theta/\partial t = C_1$. The solution of the differential equation is then:

$$(6)\quad \theta = \frac{C_1 r^2}{4} + B_1 \ln r + D_1 + C_1 t$$

where B_1 and D_1 are constants of integration; $\theta = \theta_a - \theta_b$.

Another general solution therefore might be:

$$(7)\quad \theta = \left[\frac{C_1 r^2}{4} + B_1 \ln r + D_1 + C_1 t\right] - \sum_1^\infty \left[A J_o(\mu r) + B N(\mu r)\right] e^{-\alpha\mu^2 t}$$

This equation can be used only for a limited time since the first term increases with time. The entire solution breaks down on a substitution of initial conditions. At all values of r, time equal to zero, the first term and the temperature difference $(\theta_a - \theta_b)$ are both zero. Again A equal to zero is the only choice for evaluating A. Unless θ is considered as $\theta_a - \theta_b$, the solution is merely for a condition in which all temperatures are rising at the same rate at all radii, consequently eliminating that possibility for a solution.

Assume the solution of the differential equation to be $\theta = R + T$. Differentiation and substitution results in:

$$(8)\quad \frac{1}{\alpha}\frac{\partial T}{\partial t} = \left(\frac{\partial^2 R}{\partial r^2} + \frac{1}{r}\frac{\partial R}{\partial r}\right) = -\mu^2$$

on inspection it is seen:

$$(9)\quad \frac{\partial T}{\partial t} = -\alpha\mu^2 = \frac{\partial\theta}{\partial t}$$

thus the result is the same as the solution of $\partial\theta/\partial t = C_1$.

As a result of the above investigation there does not appear, applying the above approach, to be an analytic solution to the actual problem. Through necessity, rather than through preference, an attempt was made to find a graphical solution.

GRAPHICAL SOLUTION FOR TEMPERATURE DISTRIBUTION

Schmidt's graphical method (12) is basically for a slab. It can be adapted to the cylinder, but as previously stated, it is very much in error during the early time intervals. Williamson and Adams (15) point out, and Fishenden and Saunders (5) confirm the fact, that Lord Kelvin's (8) probability integral more nearly approximates the initial thermal wave penetration. This improvement over the Schmidt method was further advanced by Perry and Berggren (10). They determine a graphical solution for the heat flow differential equation for cylinders using small finite differences. This graphical solution is developed only for the problem of a hollow cylinder with an instantaneous surface heat source. The inside wall temperature is then maintained constant. Conduction to an infinite fluid is assumed to take place at the outer wall. The results are compared with values obtained by using Carslaw's and Jaeger's formula for the same condition.

The equation given by Perry and Berggren is for the temperature rise during a time interval at any given radius (10) (6). It is merely a weighted mean value of the temperature of the two adjacent radii before the time interval, namely:

$$(10)\quad \theta_{t+\Delta t,r} = \tfrac{1}{2}\left\{\left(1-\frac{\Delta r}{2r}\right)\theta_{t,r-\Delta r} + \left((1+\frac{\Delta r}{2r}\right)\theta_{t,r+\Delta r}\right\}$$

For very small values of Δr or as $\Delta r \rightarrow 0$ the equation agrees with Schmidt's equation for a slab (12):

$$(11)\quad \theta_{t+\Delta t,r} = \tfrac{1}{2}\left\{\theta_{t,r-\Delta r}+\theta_{t,r+\Delta r}\right\}$$

The initial curve following the Lord Kelvin probability integral or Dahl's equation can be used over that first small time increment. Assume an instantaneous rise of temperature at the inner wall for the short time increment. The outer wall will remain at the initial uniform temperature. Where Δt is very small Dahl's equation has little, if any, advantage over Lord Kelvin's integral.

It is desirable to have small increments of Δr. Since Δt is proportional to Δr^2, the error of the graphical solution can be reduced to a minimum by a proper choice of Δr. By progressing through the graphical analysis using a tedious mathematical solution for each point, an accurate curve can be drawn. In the actual problem the outer surface temperature increases with the inside temperature. It is desirable to prevent a dip in the curve below the outer wall temperature. Any temperature which graphically is determined to be below the outer surface temperature will automatically be given the same temperature as the outer surface. This condition will occur during early time intervals near the outer layer. An attempt to fair in a curve would introduce possible errors on the unsafe side of the actual temperature distribution curve. It is considered better therefore to introduce a slight safety factor by using the procedure mentioned above.

Jakob (6) and Perry and Berggren (10) give proof that a scale of Δr plotted as $\Delta\lambda$, where $\lambda = \ln r/r_a$, can be used in a plot against temperature to produce a direct graphical method of solution. The temperature rise in a cylindrical segment for a time interval Δt can then be obtained merely by drawing a straight line on the graph between $\theta_{t,\lambda-\Delta\lambda}$ and $\theta_{t,\lambda+\Delta\lambda}$. The inter section $\theta_{t+\Delta t,\lambda}$ is the desired temperature.

A careful evaluation of the problem graphically yields the results as shown in Fig. 7 and Fig. 8. The tedious graphical solution represented by Fig. 7 immediately caused the author to investigate the possibility of a more rapid graphical solution. A graphical solution represented by Fig. 10 was then made for purposes of comparison.

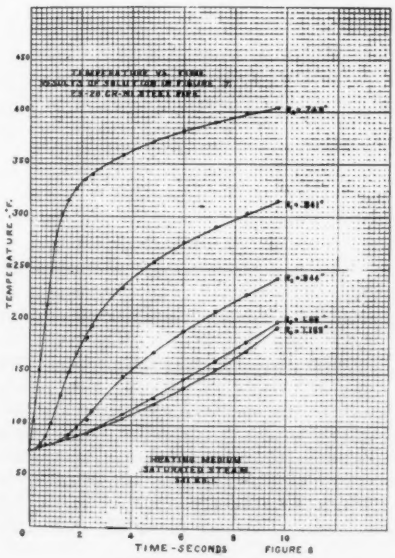
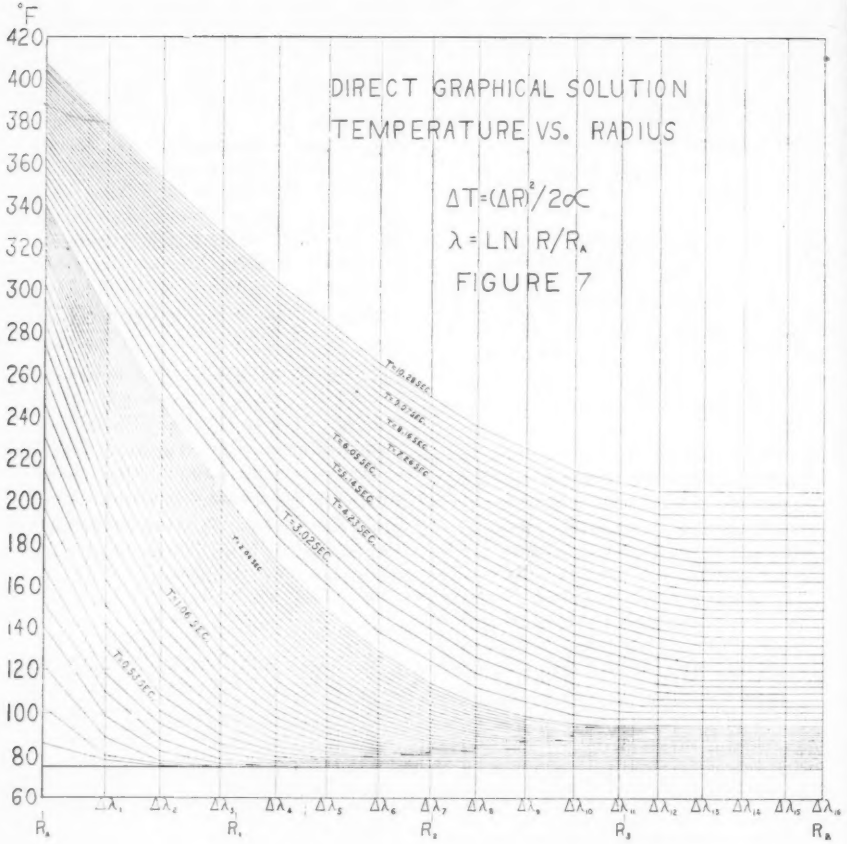
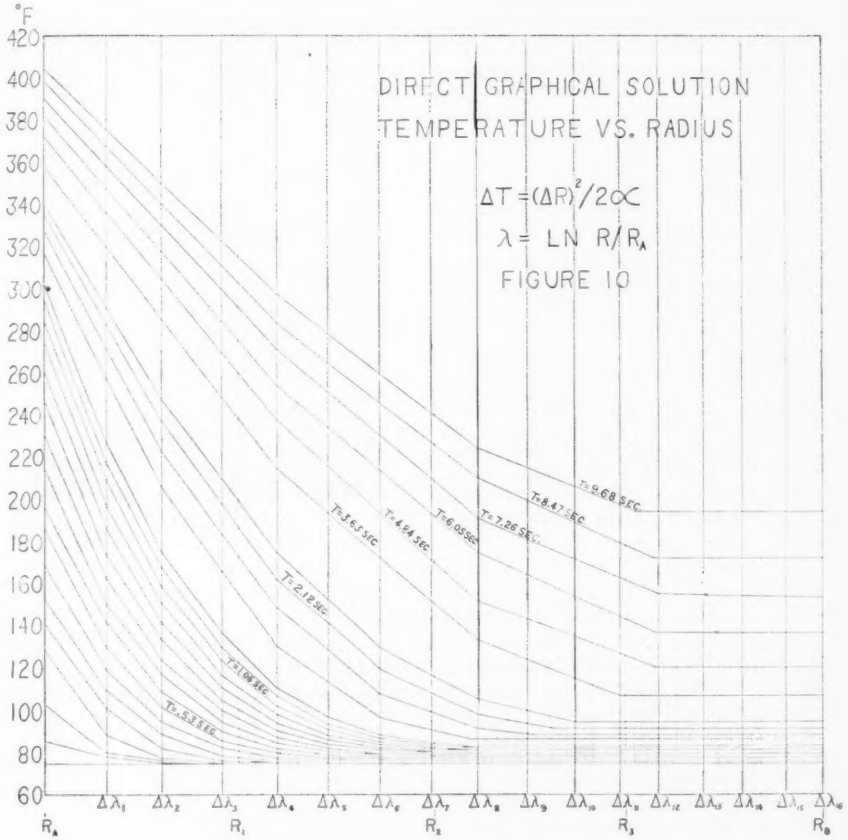


FIGURE 8



The first sixteen steps of both solutions are identical. Time increments in Fig. 10 are then rapidly increased to speed up construction. The time to produce Fig. 10 was approximately one half the time required for Fig. 7. A comparison of resultant values of temperatures for r_1, r_2 and r_3 from both Figures was then made. A maximum deviation of 6 degrees representing a 4% error occurred for r_1 at 1.512 seconds. This error rapidly decreases and disappears after 6.048 seconds. For r_2 the maximum deviation was 4 degrees for a 2.8% error at 2.1168 seconds. The trend of a decreasing deviation and an increasing time for that maximum deviation to occur was noted with an increase in radius. At r_3 the maximum deviation was 3 degrees for an error of 2.7%. The time was 3.628. The deviation disappears as time increases which is a trend characteristic of any graphical solution of this type (13).



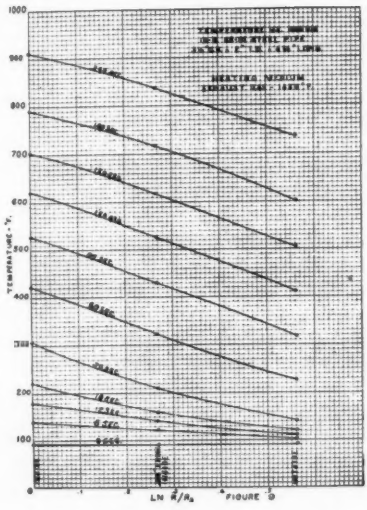
DISCREPANCIES NOTED IN DATA USED

A plot of steady state temperature distribution radially in a hollow cylinder is a straight line when plotted on coordinates of temperature vs. ln r/r_a. The data as shown in Fig. 5 represents the readings of three thermocouples in a pipe wall. This data was replotted in Fig. 9 as temperature vs. ln r/r_a.

Two peculiarities are apparent upon examination of the resultant curves. Most apparent is the change of curvature from concave up to concave down after ninety seconds. The second peculiarity at ninety seconds is a change from increasing slope with time to decreasing slope with time. The latter is the effect of the wave front as the heat penetrates the pipe wall.

The change from concave up to concave down indicates at once a discrepancy in the data. Since steady state is the ultimate condition which can exist under the conditions of the test, the concave down curves are an impossibility.

The author is of the opinion that the plot reveals a lag in temperature readings. This may be an error introduced by either the thermocouples, the indicating instrument, or both. The lag, on the other hand, may be only the result of personnel errors in taking the temperature readings. The existence of this lag must, however, be acknowledged. Pipe lagging for insulation would have no effect on the results until approximately the time at which the first heat wave reaches the outer wall, consequently the absence of lagging during these tests has no effect upon the data taken during the early stages.



CONCLUSIONS

It has become quite obvious, as a result of this investigation, that more detailed data should be taken in future tests. The installation of several thermocouples at proper radial points in the pipe walls would be desirable. The maximum temperature rate and consequently the maximum thermal shock in the test occurs within five seconds after heating begins and, in the case of severe cold shock, maximum stresses would occur in a shorter interval. Automatic recording devices attached to the thermocouples may be the answer to better data in the critical time interval. The use of oscilloscopes and cameras may be of further assistance in obtaining more accurate data.

The maximum error introduced by the rapid graphical solution is small and tends to decrease with time. Determination of the temperature distribution in a pipe wall can be made by the rapid graphical method in two to three hours. These factors may be of value to anyone who wishes to make calculations on a series of thermal shock tests for comparative purposes or as a means of checking recorded data.

It appears, therefore, that the graphical solution suggested by Perry and Berggren as applied to the actual problem is the only solution if it is desired to maintain the complex boundary conditions. It is also a solution which can be calculated with comparative ease by a direct method.

The numerical results in this paper are only as reliable as the data taken. All values of temperature prior to a time of five seconds were obtained by extrapolation. It is necessary therefore to consider the value of this paper primarily in terms of the theory presented and its application to a specific practical problem.

The author's desire to present this paper was motivated not by a feeling of having made any contribution to science regarding this particular subject but primarily from a feeling that this paper may stimulate the interest of qualified men to make a thorough investigation considering the actual boundary conditions which exist when thermal shock takes place in steam piping.

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Official Department of Defense Photo

The Douglas Ad Skyraider test fires the first successful air-to-air rocket, named the "Mighty Mouse," which has been developed by the Navy Bureau of Ordnance. The new rocket is small enough to be carried in quantity by its mother aircraft, yet fast and powerful enough to destroy any known plane with a direct hit. Built for use on modern, high-speed aircraft against other high-speed planes, the new rocket has far greater speed and range than the aircraft rockets developed during the last war.

NICKEL-MOLYBDENUM-VANADIUM ALLOY STEEL SHIELDED ARC WELDING ELECTRODES (LOW HYDROGEN TYPE)

E. H. FRANKS, C. T. GAYLEY, W. H. WOODING

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MR. C. T. GAYLEY received the degree of B.S. in Chemistry at West Virginia University in 1917. Worked for Westinghouse Electric Company on metallurgical problems concerned with the development of welding and brazing processes, fluxes and arc welding electrodes. Appointed as civilian metallurgist at Industrial Test Laboratory of the Philadelphia Naval Shipyard in 1937. Reappointed Welding Engineer, 1940, from which time he has been engaged in investigations concerning welding metallurgy, welding processes and the development of welding electrodes. He is a member of the American Welding Society and has been author and coauthor of a number of technical papers.

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INTRODUCTION

During the past fifteen years, the Navy has enjoyed wide experience in the shielded arc welding of Special Treatment Steel in ship construction. Most of this experience has been gained through the use of the Navy Grade IV (25-20 Chrome-Nickel) austenitic steel welding electrode covered by Bureau of Ships Specification 46E4(INT). The resulting weld deposits, though considered satisfactory, were generally plagued with small intergranular cracks and the joint efficiencies obtained approximated 70 percent of the plate material.

At the beginning of World War II, the War Production Board, in attempting to conserve strategic materials, prohibited the use of the Navy Grade IV electrode and directed that all armor welds be made using the Navy Grade V (20-10 Chrome-Nickel) austenitic steel welding electrode. Most everyone associated with shipbuilding is familiar with the difficulties encountered using this substitute electrode. Weld metal cracking along the line of fusion was so prevalent that it was practically impossible to obtain satisfactory butt welds under the conditions of high restraint normally encountered in shipbuilding. And, where occasionally butt weld joints were babied to a satisfactory completion, sudden drops in atmospheric temperature left little to brag about. Extensive fusion line cracking has been known to develop with sudden drop in atmospheric temperature to below freezing occurring as late as thirty days after completion of a supposedly satisfactory butt weld.

It was because of these difficulties that the War Production Board relaxed their ruling to permit the use of Navy Grade IV (25-20) austenitic steel welding electrodes in depositing multipass butt welds in 20-pound or heavier plate. Consequently, the National Defense Research Council, following through with W.P.B.'s efforts to conserve strategic materials, entered upon a program to develop a ferritic type armor welding electrode. This electrode, commonly called NRC-2a, was of the manganese-molybdenum alloy steel type and its covering formulation approximated that of the Navy Grade IV (25-20) electrode.

During the same period, shipbuilders also were experiencing difficulty welding Navy High Tensile Steel using the AWS-E-6010 mild steel welding electrode (Grade 1 of Bureau of Ships Specification 46E3). Since this trouble, underweld cracking, could be greatly reduced using an austenitic type (Grade IV), welding electrode, the Navy also found itself interested in finding a ferritic type electrode which would overcome this difficulty. These efforts resulted in the development of a molybdenum alloy type ferritic electrode having a covering formulation similar to that of the austenitic type and which was commonly referred to as a "Mineral Covered High Tensile Steel Welding Electrode." It is doubtful that anyone ever thought the troubles encountered using the Navy Grade V electrode would ever be dwarfed, but the NRC-2a and the high tensile steel electrodes were found to be so erratic and unpredictable in their behavior that they succeeded in overshadowing the previous troubles and it was necessary to withdraw them for further study and development. These studies resulted in the startling discovery that, for the most part, the erratic behavior of these electrodes was caused by moisture in the covering; both that absorbed from the atmosphere and through the use of compounds in the covering having a high content of water of crystallization. It was this information concerning the effect of moisture that has been largely responsible for the success of the current low hydrogen type AWS-E-7015 medium steel welding electrode covered by Bureau of Ships Specification 46E8(INT).

With this background, the Navy in cooperation with NEMA, the National Electrical Manufacturers Association, of which the electrode manufacturers are members, formed a committee of technical personnel from both the electrode industry and the Navy, to develop an electrode that would deposit high-yield, notch-tough, ferrous weld metal. This program was to be carried on in conjunction with that for the development of high-yield, notch-tough steels for hull construction. The electrode to be developed was to have operating characteristics similar and equal to those of the low hydrogen type electrode. The deposited weld metal was to have a yield strength of from 85,000 to 95,000 p.s.i. and a ductility of approximately 20 percent elongation in 2° test lengths. The physical properties of the weld deposit were to overmatch the plate material by approximately 10 percent. The electrode was to be suitable for depositing satisfactory welds at preheat and interpass temperatures of 0°F; and, under the highest degree of restraint that could be successfully welded using AWS-E-7015 or 7016 electrodes under like conditions.

It was further agreed that the impact properties of the weld deposit should indicate a transition temperature of less than —40°F. Other than the fact that the weld deposit was to be ferritic, no restrictions were placed on the type and quality of alloy to be introduced into the weld deposit.

Of course, a modification of the NRC-2a electrode was the first to be investigated in this program. The electrode is of the manganese-molybdenum type and its deposited weld metal meets the yield strength, elongation and impact test requirements given above. However, it was found that satisfactory multipass welds could not be deposited at preheat and interpass temperatures of less than 150°F, even under very moderate degrees of restraint. This indicated that preheat and not restraint was the primary consideration for successful application of the electrode.

The development program proceeded with the investigation of many types of electrodes having various combinations of alloying elements. These included:

	Nominal Alloy Composition
	per- cent *per- cent* *per- cent*
Manganese-Molybdenum	(1.7 - 0.30)
Manganese-Chromium	(0.90 - 0.90)
Manganese-Vanadium	(1.0 - 0.20)
Molybdenum	(1.0)
Molybdenum-Vanadium	(1.0 - 0.10)
Molybdenum-Nickel-Chromium	(0.80 - 0.80 - 0.30)
Nickel-Chromium	(1.90 - 0.80)
Nickel-Chromium-Vanadium	(1.90 - 0.30 - 0.20)
Nickel-Manganese	(1.80 - 1.30)
Nickel-Manganese-Vanadium	(2.0 - 1.0 - 0.15)
Nickel-Silicon	(1.6 - 0.75)
Nickel-Molybdenum	(1.6 - 0.50)
Nickel-Molybdenum-Vanadium	(1.6 - 0.35 - 0.10)

The weld deposits for most of these alloy combinations met the tensile property requirements of the program, but fell very short in meeting the weldability requirements. This caused considerable concern among the committee members and, as a result, many coating formulations were investigated for a number of the above alloy compositions, greatly increasing the scope of the investigation. However, it was not until the nickel-molybdenum-vanadium alloy type had been investigated that an electrode was found which successfully met the weldability requirements; that is, depositing satisfactory weld metal under a high degree of restraint at preheat and interpass temperatures of 0°F. Further work with this alloy composition indicated that

only one manufacturer could produce a satisfactory electrode of this type. In view of this, the program was again enlarged to permit a study of various coating formulations in an endeavor to determine the cause for the inability of other electrodes of this type to perform satisfactorily. This study indicated that, for the most part, the moisture content of the covering greatly influenced the performance of the electrode and that baking at high temperatures to reduce the moisture content to a value much lower than that previously found necessary was of considerable benefit.

At that time, the Bureau of Ships became interested in obtaining a ferritic type alloy steel electrode to take the place of the present Navy Grade IV (25-20) austenitic electrode currently used for welding Special Treatment Steel. The weld deposit was to have a minimum yield strength of 110,000 p.s.i. and good low temperature impact properties. A further requirement was that the electrode should be suitable for use under conditions of very high restraint at preheat and interpass temperatures of 70 to 100°F. In view of the success experienced in using the nickel-molybdenum-vanadium type composition for welding the high-yield strength steels, work proceeded along this line of attack, resulting in a rather rapid development of the present electrode. This electrode is similar in all respects to the 90,000 p.s.i. type except that the molybdenum content has been increased to 0.75 percent and the vanadium to 0.20 percent. However, the impact properties of the restrained weld deposits are not quite as good as had been anticipated. Despite the high strength of the weld deposit, the handling and operating characteristics of this electrode are quite similar to those of the 90,000 p.s.i. minimum yield and the AWS-E-7015 or 7016 electrodes.

Being well cognizant of that famous Welder Philosophy "Believe not what you hear and only 50 percent of what you see provided you can X-ray"; it was deemed advisable to send samples of this electrode to various shipbuilding activities to obtain their comments concerning its performance. Quantities were shipped to various Naval Shipyards and to several east coast private shipbuilding companies. Without exception the comments which have been received from these activities have been most encouraging. In some instances reference was made to minor difficulties encountered in depositing groove welds such as porosity at the start of the weld, root bead cracking and crater cracks. However, in these instances the testing agencies reported that such difficulties were easily overcome by slight changes in technique.

In view of the many favorable comments, the electrode was recommended for use in the fabrication of a large pressure vessel. It was considered that this structure would provide an excellent proving ground for the electrode. The reaction of the welders concerning the use of the electrode was most complimentary. After one week of welding, most of them were outspoken in their preference of this electrode over the Navy Grade IV (25-20) austenitic type. However, this activity did encounter one difficulty not previously reported and that was transverse cracking in the reinforcement beads adjacent to the base metal.

Though the usability and handling characteristics of the electrode and its favorable reception by various shipbuilding yards are most gratifying, it is conceded that there is still room for further improvement of the physical properties insofar as impact is concerned. Such improvement probably will necessitate changes in the alloy balance of the deposit and will not necessarily require any major alteration in the coating formulation. Consequently, any improvement in physical properties of the weld deposit should have no appreciable

effect upon the usability characteristics of the electrode.

The typical chemical and physical properties of the Ni-Mo-V electrode and its deposit are shown in Table No. 1.

As shown in Table No. 1, the ingredients used in the manufacture of the covering indicate it to be a lime-titania type to which has been added various alloying elements. Of particular note is the absence of organic or cellulosic materials. Such electrodes are generally referred to as the "Low Hydrogen Type" in that the arc atmosphere produced by such coverings are low in hydrogen content. The composition of the core wire indicates it to be a rimmed mild steel. It has a very low silicon content and except for manganese such alloying elements as are present might well be considered impurities. However, the composition of the weld deposit shows considerable alloy content which has been accomplished by transfer of the alloying ingredients in the covering. This alloy recovery approximates 75 percent of that present in the covering. It is improbable that production of these electrodes using alloy steel core wire would prove more economical; but this phase of the investigation has not as yet been completed.

The physical properties of the weld deposit are of particular interest. The

TABLE 1

TYPICAL CHEMICAL AND PHYSICAL PROPERTIES
OF
NICKEL-MOLYBDENUM-VANADIUM ALLOY STEEL
SHIELDED ARC WELDING ELECTRODE

CHEMICAL PROPERTIES

Covering Ingredients	*Core Wire*		*Weld Metal**	
		percent		*percent*
Limestone	Carbon	0.10	Carbon	0.07
Fluorspar	Manganese	0.39	Manganese	0.67
Titania	Phosphorus	0.010	Phosphorus	0.018
Ferro-Manganese	Sulphur	0.023	Sulphur	0.023
Ferro-Silicon	Silicon	0.01	Silicon	0.37
Nickel	Nickel	0.13	Nickel	1.79
Ferro-Molybdenum	Chromium	0.06	Chromium	0.06
Ferro-Vanadium	Molybdenum	0.01	Molybdenum	0.79
Sodium Silicate	Vanadium	0.01	Vanadium	0.28

* All Weld Metal Pad Analysis.

PHYSICAL PROPERTIES

(ALL WELD METAL — CLAD SCARFS)

TENSILE PROPERTIES (.505" Spec.)		IMPACT PROPERTIES (Charpy Keyhole)	
			Ft. Lbs.
Yield Strength	116,000	70°F	33.8
Tensile Strength	123,000	0°F	27.2
Elongation in 2"	20.8%	—40°F	23.8
Reduction in Area	54.8%	—60°F	17.2

yield and tensile strengths of the deposited weld metal are in excess of the desired minimum with exceptional elongation and reduction in area for high strength, cast steel. Most tensile fractures show a full cup similar to that generally observed in forged alloy steel specimens. The impact properties indicate fairly good energy absorption with a transition temperature somewhat lower than —40°F. However, impact properties of restrained groove weld metal are somewhat disappointing in that they fall off rapidly at low temperatures and indicate a transition temperature approximating 0°F. Present development is continuing in an effort to improve these impact properties.

OPERATING CHARACTERISTICS

Figs. 1 through 4 compare some of the operating characteristics of the nickel-molybdenum-vanadium electrode with those of the Navy Grade IV (25-20) austenitic steel electrode. As would be expected, Fig. 1 shows the burn-off rate of the 3/16" diameter Ni-Mo-V type electrode to be more rapid in comparison to that of the same size 25-20 electrode when both are operated at their optimum currents of 225 amperes and 175 amperes respectively. At the same welding current, the burn-off rate of the 3/16" diameter Ni-Mo-V type electrode is somewhat less than that of the 25-20 type electrode. However, as the current is increased, the metal transfer characteristics of the Ni-Mo-V type electrode improve considerably and approach the ideal spray effect at about 210 amperes. The 25-20 type electrode not only fails to show this improvement with increase in current, but at the higher currents becomes excessively overheated.

The deposition rates shown in Fig. 2 indicate that the 25-20 electrode at 175 amperes is about 30 percent less efficient in weight of weld metal deposited than the comparable Ni-Mo-V type electrode operating at 225 amperes. Even at the optimum current for 25-20 electrodes the deposition efficiency of the Ni-Mo-V electrode compares favorably with the austenitic type.

Fig. 3 shows the depth of penetration as a function of welding current for Ni-Mo-V and 25-20 electrodes. As might be expected, penetration is a direct function of the welding current when the arc voltage is held constant at 23 arc volts over the range. At ap-

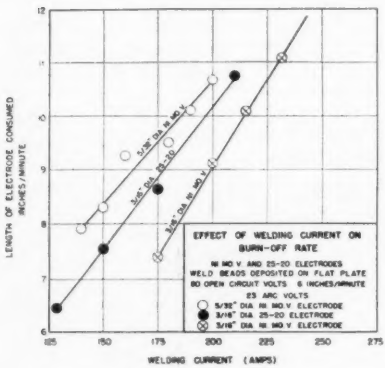

LENGTH OF ELECTRODE CONSUMED INCHES/MINUTE
WELDING CURRENT (AMPS)
EFFECT OF WELDING CURRENT ON BURN-OFF RATE
Ni MO V AND 25-20 ELECTRODES
WELD BEADS DEPOSITED ON FLAT PLATE
80 OPEN CIRCUIT VOLTS 6 INCHES/MINUTE
23 ARC VOLTS
5/32" DIA Ni MO V ELECTRODE
3/16" DIA 25-20 ELECTRODE
3/16" DIA Ni MO V ELECTRODE

Figure 1

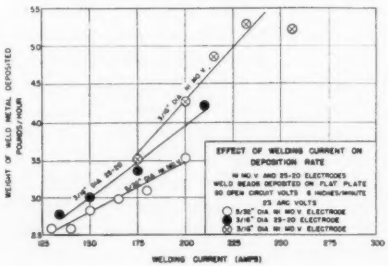

WEIGHT OF WELD METAL DEPOSITED POUNDS/HOUR
WELDING CURRENT (AMPS)
EFFECT OF WELDING CURRENT ON DEPOSITION RATE
Ni MO V AND 25-20 ELECTRODES
WELD BEADS DEPOSITED ON FLAT PLATE
80 OPEN CIRCUIT VOLTS 6 INCHES/MINUTE
23 ARC VOLTS
5/32" DIA Ni MO V ELECTRODE
3/16" DIA 25-20 ELECTRODE
3/16" DIA Ni MO V ELECTRODE

Figure 2

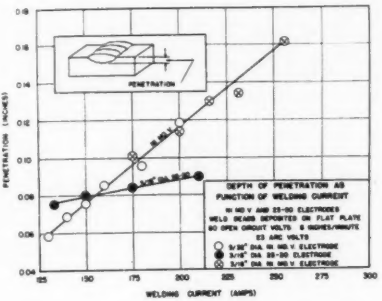


Figure 3

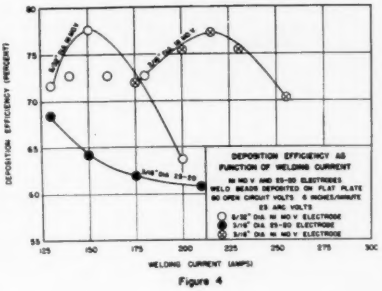
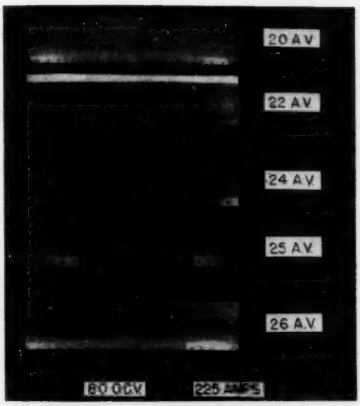


Figure 4

proximately 155 amperes both the Ni-Mo-V and the 25-20 electrodes exhibit the same depth of penetration. However, the 25-20 electrode shows little increase in penetration with increased currents, while the penetration for the Ni-Mo-V type shows significant increase with higher currents. It is interesting to note that the penetration for both the 5/32″ and 3/16″ diameter Ni-Mo-V electrodes follow the same curve over their respective current ranges.

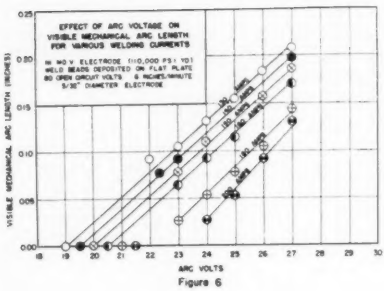
The deposition efficiencies of Ni-Mo-V type and 25-20 type electrodes at several welding currents are presented in Fig. 4. As indicated, a maximum efficiency of approximately 77 percent occurs at 150 and 215 amperes for the 5/32″ and 3/16″ diameter Ni-Mo-V type electrodes, respectively. At higher currents, the spatter loss reduces the efficiency considerably. Why this efficiency falls off at the lower currents is not readily understood. The deposition efficiency of 3/16″ diameter 25-20 electrodes operating at 175 amperes approximates 62 percent which is considerably lower than that obtained for the Ni-Mo-V electrode. The data shown in Fig. 4 indicate the optimum current range for the Ni-Mo-V electrodes to be 140-160 amperes for 5/32″ diameter and 200-230 amperes for the 3/16″ diameter.

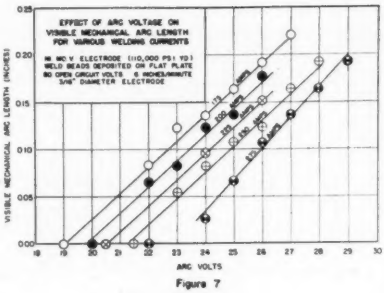
Having demonstrated the effect of welding current on the operating characteristics of the Ni-Mo-V type electrode, it appeared desirable to establish the effect of arc voltage at the optimum current. The results of these determinations are shown in the exographs presented in Fig. 5 and show the radiographic characteristics of deep groove welds, ¾″ thick by 12″ long, deposited in standard 0.505″ tensile test assemblies using automatic voltage controlled arc welding equipment. Operating at the optimum current of 225 amperes for 3/16″ diameter electrodes, the welds deposited at 20 and 22 arc volts appear to be X-ray free of porosity. At 24 arc



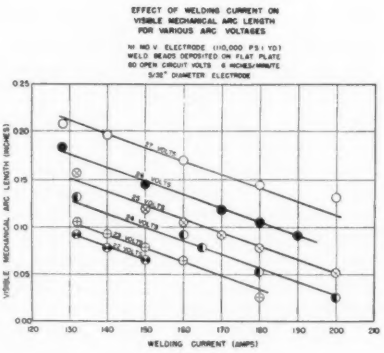
EXOGRAPHS OF DEEP GROOVE WELDS SHOWING EFFECT OF ARC VOLTAGE

Figure 5

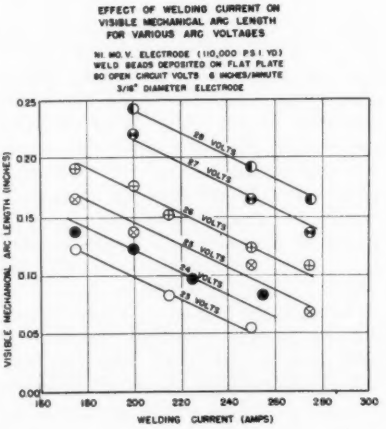

EFFECT OF ARC VOLTAGE ON
VISIBLE MECHANICAL ARC LENGTH
FOR VARIOUS WELDING CURRENTS
VISIBLE MECHANICAL ARC LENGTH (INCHES)
ARC VOLTS
Figure 6


EFFECT OF ARC VOLTAGE ON
VISIBLE MECHANICAL ARC LENGTH
FOR VARIOUS WELDING CURRENTS
VISIBLE MECHANICAL ARC LENGTH (INCHES)
ARC VOLTS
Figure 7

volts a few slag stringers appear. At 25 arc volts, the exograph indicates the presence of some porosity and macrographic examination of transverse sections through this weld indicate that the major portion of this porosity was located in the sealing or reinforcing layer. However, at 26 arc volts, there was considerable porosity throughout the deposit. These data would indicate that 24 arc volts appear to be the maximum for satisfactory operation of 3/16" diameter Ni-Mo-V electrodes. In view of these observations, it was considered desirable to establish the maximum visible mechanical arc length for satisfactory operation and its relationship to change in arc voltage and welding current.


EFFECT OF WELDING CURRENT ON
VISIBLE MECHANICAL ARC LENGTH
FOR VARIOUS ARC VOLTAGES
VISIBLE MECHANICAL ARC LENGTH (INCHES)
WELDING CURRENT (AMPS)
Figure 8

Figs. 6 and 7 show the visible mechanical arc length as a function of arc voltage at various welding currents. As is to be expected, the visible mechanical arc length is a direct function of the arc voltage at any one current value in that, as the voltage is increased, so does the mechanical arc length. The zero visible mechanical arc lengths are possible with the Ni-Mo-V electrode, it being practically impossible to short out the arc without intentionally breaking the covering forming the crucible. However, the variation of current also produces a family of straight lines, with increasing welding currents showing a shorter visible mechanical arc length at any given arc voltage. This is shown in Figs. 8 and 9, which present the effect


EFFECT OF WELDING CURRENT ON
VISIBLE MECHANICAL ARC LENGTH
FOR VARIOUS ARC VOLTAGES
VISIBLE MECHANICAL ARC LENGTH (INCHES)
WELDING CURRENT (AMPS)
Figure 9

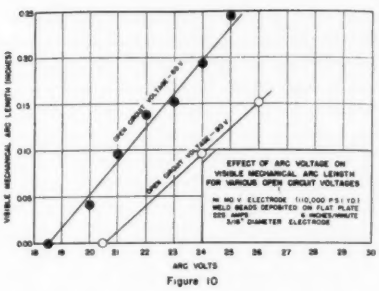

VISIBLE MECHANICAL ARC LENGTH (INCHES)
ARC VOLTS
EFFECT OF ARC VOLTAGE ON VISIBLE MECHANICAL ARC LENGTH FOR VARIOUS OPEN CIRCUIT VOLTAGES
OPEN CIRCUIT VOLTAGE = 80 V
OPEN CIRCUIT VOLTAGE = 60 V

Figure 10

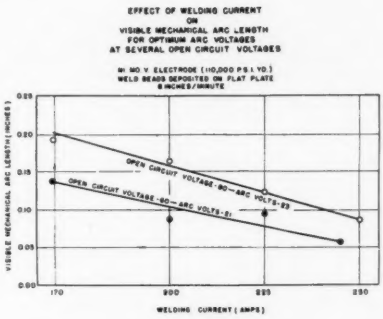

EFFECT OF WELDING CURRENT ON VISIBLE MECHANICAL ARC LENGTH FOR OPTIMUM ARC VOLTAGES AT SEVERAL OPEN CIRCUIT VOLTAGES
OPEN CIRCUIT VOLTAGE-80—ARC VOLTS-25
OPEN CIRCUIT VOLTAGE-60—ARC VOLTS-21
VISIBLE MECHANICAL ARC LENGTH (INCHES)
WELDING CURRENT (AMPS)

Figure 11

of welding current on visible mechanical arc length for various arc voltages. Here it can be seen that the visible mechanical arc length at a constant arc voltage diminishes as the welding current is increased. These data in conjunction with that of Fig. 5 appear to establish a maximum visible mechanical arc length of 0.10 inch for satisfactory operation of the 3/16″ diameter electrode. Though not further investigated by radiographic studies, it is believed that this value of visible mechanical arc length is also applicable to the 5/32″ diameter electrode and probably to all sizes.

Fig. 10 shows the visible mechanical arc length as a function of arc voltage for open circuit voltages of 80 and 60. For 80 open circuit volts and 225 amperes, the visible mechanical arc length at 24 arc volts is approximately 0.10″ and diminishes as a straight line function with reduction in voltage to 0.00″ at 20.5 arc volts. Again considering the weld assemblies presented in Fig. 5, satisfactory welds were deposited between 24 and 20.5 arc volts or over a range of visible mechanical arc lengths of 0.10″ to 0.00″ when the open circuit voltage was maintained at 80 volts. A reduction in the open circuit voltage to 60 would require a reduction in the arc voltage range to that of 21 to 18.5 arc volts if the electrode is to be operated within the satisfactory range of visible mechanical arc length.

The effect of welding current on the visible mechanical arc length of 3/16″ diameter Ni-Mo-V electrodes at open circuit voltages of 80 and 60 is presented in Fig. 11. As shown, the arc length increases with decrease in welding current and the relationship is the same for both open circuit voltages.

Fig. 12 shows the depth of crucible formed at the end of the 3/16″ and 5/32″ diameter Ni-Mo-V electrodes as a function of welding current. The data show that the crucible depth is a direct function of welding current. It is because of the variation in the depth of the crucible as a function of current that the arc voltage must be reduced at the lower currents to keep the visible mechanical arc length within the satisfactory operating range. It is interesting to note that the crucible depth is in direct relationship to welding current for the two sizes of electrodes.

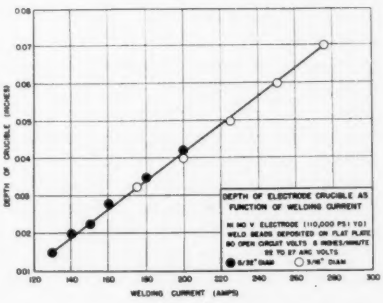
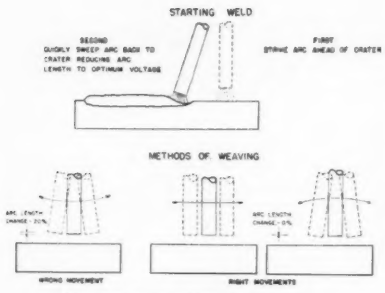


Figure 12



POINTERS FOR ELECTRODE MANIPULATION

Figure 13

Based upon experience and observation of the operation of the Ni-Mo-V type electrode, Fig. 13 incorporates a few instructions for manipulation of this electrode. In continuing a weld bead, it is not desirable to strike the arc in the crater of the previous deposit since momentary lengthening of the arc usually occurs upon striking. As previously indicated the lengthening of the arc may result in porosity at the start of the weld deposit. A better method is to strike the arc ahead of the weld, quickly bringing same back into the crater and rapidly reducing arc length to a minimum. In weaving, the electrode should not be pivoted at the wrist as this will result in a change of arc length. Weaving should be in a straight line or pivoted below the arc so that no change in arc length results. In any event, it is advisable to restrict weave technique to a transverse oscillation of not more than two electrode covering diameters if unintentional arc lengthening is to be avoided.

EFFECT OF MOISTURE

As previously mentioned, the moisture content of the electrode covering has a considerable influence on the weld deposit. The effect of various moisture contents upon the performance of Ni-Mo-V electrodes is presented in Fig. 14. These various experimental batches of Ni-Mo-V electrodes were investigated in the Navy Circular Patch Test. The hatched columns indicate unsatisfactory performance while the solid columns show satisfactory performance and the moisture content for each condition is shown by the height of the column. As a result of this work, a maximum limit of 0.20 percent moisture content was established for satisfactory operation of these electrodes. Though no attempt is made to explain the mechanism involved, the results obtained clearly indicated the effect of moisture. Unsatisfactory performance was indicated by severe weld metal cracking while satisfactory assemblies were radiographically free of weld metal cracks.

These data concerning the effect of moisture raise the question concerning the susceptibility of this type of electrodes to the reabsorption of moisture when exposed to moisture-laden atmosphere. Factual information concerning the behavior of the Ni-Mo-V electrode when exposed for extended periods of time to high relative humidity is not at

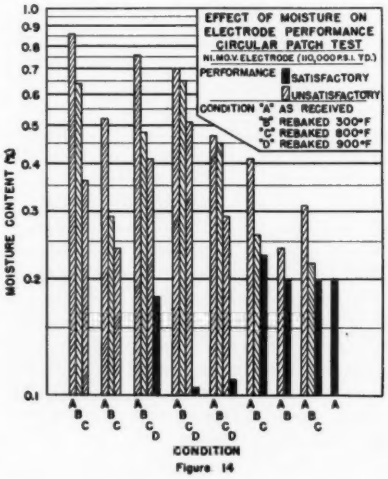
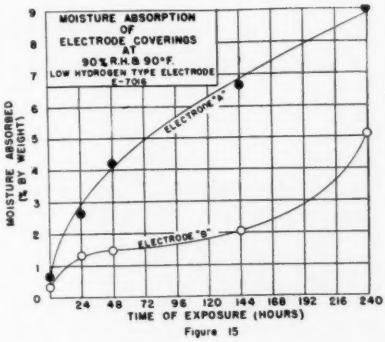


Figure 14

present available. However, data concerning the moisture reabsorption characteristics of the companion or prototype electrode AWS-E-7016 are presented in Fig. 15 for two different lots of this type electrode. When exposed openly to 90 percent relative humidity at 90°F., both lots of electrodes quickly absorb moisture at such a rate that the water content of the coverings exceed 1 percent within twelve hours. As indicated, coverings may have different moisture absorption rates over extended periods of exposure. However, the question which is not answered is how long can an electrode be exposed to these conditions before the moisture content of the covering exceeds the maximum for satisfactory operation. For openly exposed electrodes, the time would be very short and probably not more than an hour or so. However, closely packed in an open container, they would probably withstand such conditions for a period of four or five hours.

Having absorbed moisture, the next question is how to get rid of it. This is shown in Fig. 16 which again presents data on the AWS-E-7016 electrode which is the prototype for the Ni-Mo-V electrode. It is to be admitted that the moisture content that can be tolerated in the AWS-E-7016 electrode coverings approximates 0.8 percent and that the values presented in Fig. 16 represent those obtained on additional baking at the times and temperatures indicated. However, the information presented in Fig. 14 indicates that comparable trends might be expected for the Ni-Mo-V type electrode coverings.

These observations show the effect of time and temperature in reducing the residual moisture content of electrode coverings. As usual, some observations go astray and, as can be seen, the one hour determinations show considerable scatter in comparison to the others. So liberty has been taken in plotting the one hour curve as a broken line in a position relative to the other three curves. The data here would indicate


MOISTURE ABSORPTION OF ELECTRODE COVERINGS AT 90% R.H. & 90°F.
LOW HYDROGEN TYPE ELECTRODE E-7016
ELECTRODE "A"
ELECTRODE "B"
MOISTURE ABSORBED (% BY WEIGHT)
0 1 2 3 4 5 6 7 8 9
0 24 48 72 96 120 144 168 192 216 240
TIME OF EXPOSURE (HOURS)
Figure 15

that after one hour or more at temperature, the degree of temperature has a greater effect in reducing moisture than does time at temperature. It also indicates that temperatures as high as 900°F. may be required to effectively reduce the moisture below the maximum of 0.20 percent. Cognizance of this fact has been taken in the manufacture of Ni-Mo-V electrodes and they are now being baked at from 800 to 900°F. before packing by the manufacturer. After once having been baked at this temperature and not subsequently overexposed to high humidity, it is believed that rebaking at 500°F for one hour will be sufficient to recondition the electrode. However, where they have been overly exposed to high humidity, it may be necessary to rebake at 900° F for satisfactory reconditioning. In any event, it is not considered desirable to exceed a baking temperature of 900° F because of the possibility of oxidizing the finely divided ferro alloys in the covering.

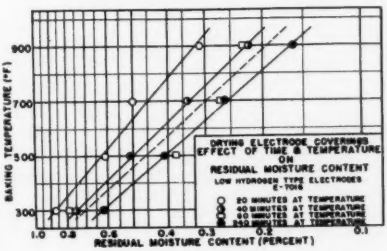


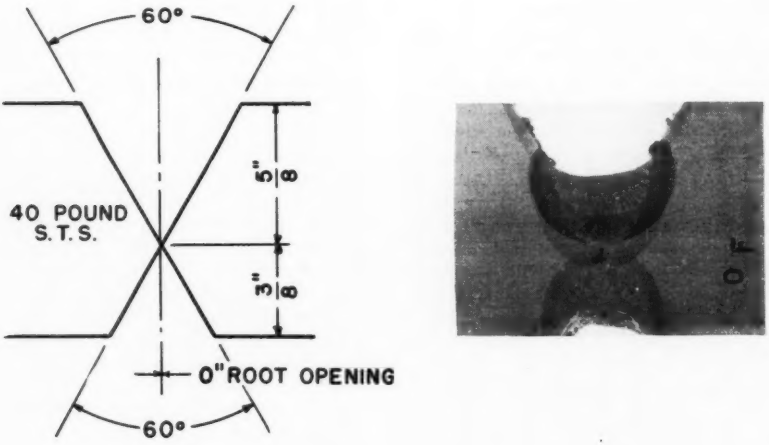
Figure 16

CONSTRUCTION PROBLEMS

Now a word concerning some of the difficulties encountered using the Ni-Mo-V electrode in the field. First, there was a problem of root cracking and burn-through in double "V" butt joints.

A subsequent investigation concerning the effect of root opening on this condition is presented in Figures 17 to 21 inclusive. Figure 17 shows a macro of a transverse section through the root layers of a double vee butt joint having a zero root opening. It is to be noted that the use of high welding currents with "O" root opening resulted in incomplete penetration at the root. Figures 18, 19 and 20 show that where root spacing is maintained not in excess of 5/32"; which is equal to the diameter of the electrodes used, full penetration at the root of the weld can be obtained. However, as shown in Figure 21, where the root spacing exceeds the diameter of the electrode, considerable burn-through and cracking in the deposit are encountered. This would indicate the necessity for rigid regulation of root spacing of at least 3/32" and not greater than that of the diameter of the electrode to be used. Consequently, good fit-up is a prerequisite if satisfactory butt welds are to be obtained with the Ni-Mo-V type electrode.

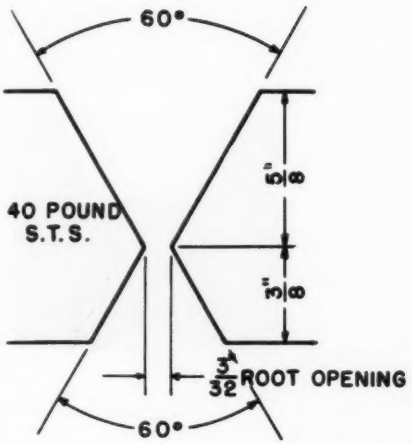
Another difficulty encountered was cracking of the base metal adjacent to the root pass of double "V" butt welds in Special Treatment Steel.

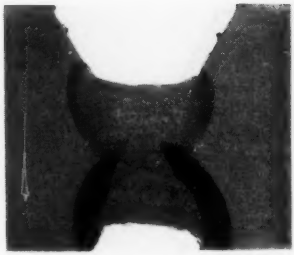


FLAT POSITION
5/32" DIAMETER ELECTRODE
180 AMPERES

ROOT PENETRATION IN DOUBLE "V" GROOVE WELDS

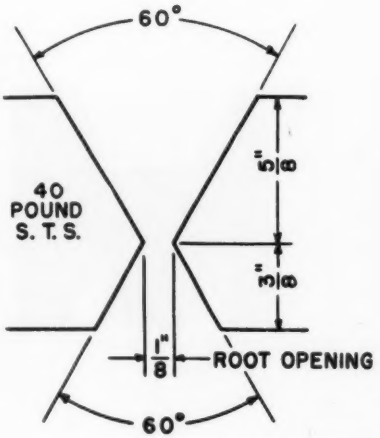
Figure 17

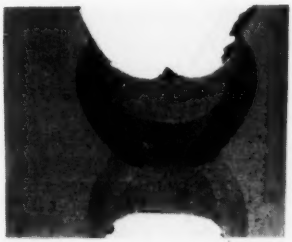




FLAT POSITION
5/32" DIAMETER ELECTRODE
160 AMPERES
ROOT PENETRATION IN DOUBLE "V" GROOVE WELDS

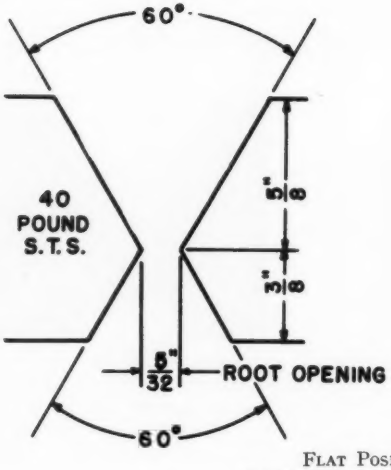
Figure 18

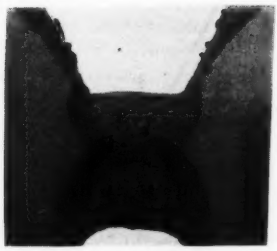




FLAT POSITION
5/32" DIAMETER ELECTRODE
136 AMPERES
ROOT PENETRATION IN DOUBLE "V" GROOVE WELDS

Figure 19

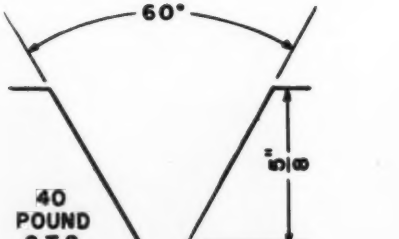




Flat Position
5/32" Diameter Electrode
130 Amperes

Root Penetration in Double "V" Groove Welds

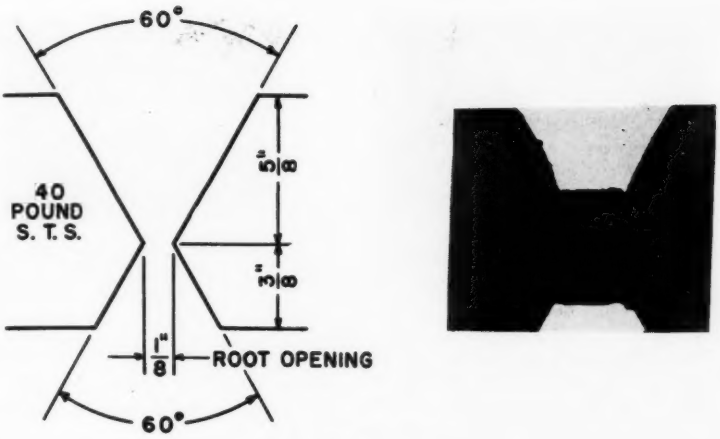
Figure 20



Flat Position
5/32" Diameter Electrode
130 Amperes

Root Penetration in Double "V" Groove Welds

Figure 21



FLAT POSITION
5/32" DIAMETER ELECTRODE
136 AMPERES
SCARF CRACKING IN DOUBLE "V" GROOVE WELDS

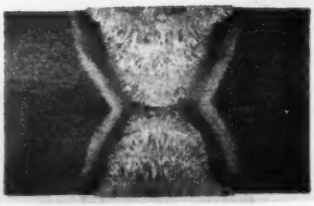
Figure 22

Figure 22 is a macrograph showing one of these cracks. Attention is invited to the slight convexity of the bead forming sharp reentrant angles at the scarf; and, as illustrated, the crack appears to originate at the apex of the angle and progress downward into the base metal. Not only are such sharp notches potential crack starters but they also form voids for slag entrapment. Therefore, it is essential that the root beads approach some degree of concavity by being deposited so that they wash up on the sides of the scarf to prevent the formation of these reentrant angles if such difficulties are to be avoided.

A further means of avoiding difficulties in the root of such welds has been accomplished through the use of AWS-E-7015 or E-7016 electrodes in the deposition of the first root pass. Figures 23 and 24 are macrographs of completed welds in special treatment steel; the former being deposited with the Ni-Mo-V type electrode throughout and the latter being deposited by using an E-7015 electrode for the first or root pass with the remainder deposited with the Ni-Mo-V type electrode.

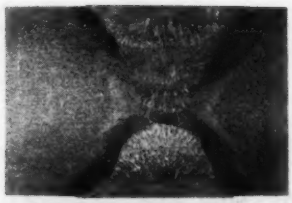
The tensile properties determined for these assemblies using full thickness strap type tensile specimens are as follows:

	Tensile Strength P.S.I.	*Yield Strength P.S.I.*	*Elongation % in 8"*	*Location of Fracture*
Figure 23	117,750	101,250	14.8	Base Metal
Figure 24	120,000	102,100	14.7	Base Metal
Navy Grade IV	99,500	71,050	2.17	Weld Metal



MACROGRAPH OF TRANSVERSE SECTION
DOUBLE "V" GROOVE WELD
ALL LAYERS DEPOSITED WITH NI. MO. V.
ALLOY STEEL WELDING ELECTRODE
MAGNIFICATION 4X

Figure 23

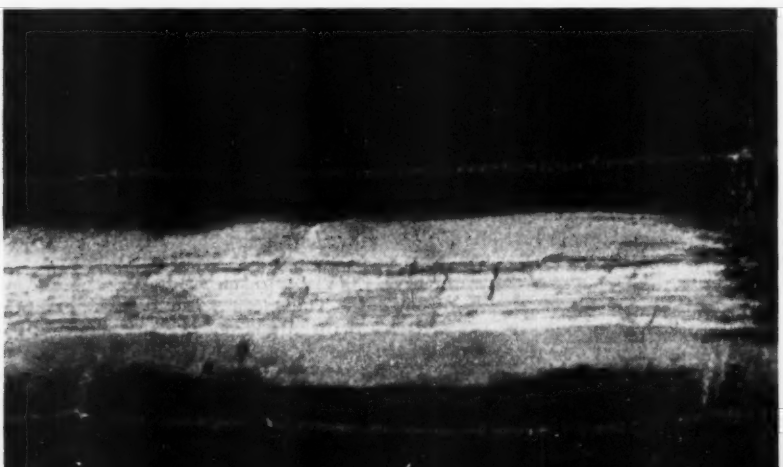


MACROGRAPH OF TRANSVERSE SECTION
DOUBLE "V" GROOVE WELD
INITIAL ROOT LAYER DEPOSITED WITH AWS E-7015 ELECTRODE
REMAINDER DEPOSITED WITH NI. MO. V. ELECTRODE
MAGNIFICATION 4X

Figure 24

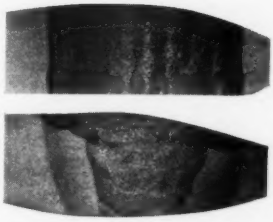
The above values are an average obtained for four specimens from each assembly. The Navy Grade IV (25-20) austenitic weld deposit values were obtained from the same type of assembly and are presented for information only.

Figure 25 is a photograph of several feet of butt weld in the pressure vessel previously referred to. The photograph shows several magnaflux indications of reinforcement layer cracks in this length of butt. These cracks did not occur with any degree of regularity but were found in groups, unevenly spaced along the weld.



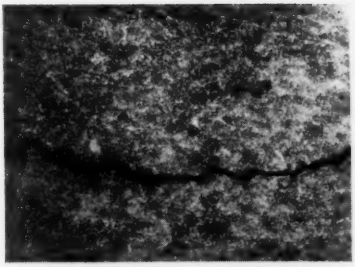
CRACKS REVEALED BY MAGNAFLUX REINFORCEMENT LAYER OF DOUBLE "V" GROOVE WELD

Figure 25



MACROGRAPHS SHOWING LOCATION OF CRACKS
DOUBLE "V" GROOVE WELD
MAGNIFICATION 2X

Figure 26



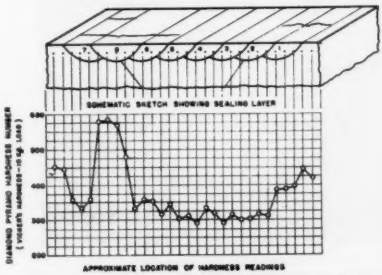
PHOTOMICROGRAPH OF CRACK IN REINFORCEMENT LAYER
DOUBLE "V" GROOVE WELD
MAGNIFICATION 200X

Figure 27

Figure 26 presents macrographs taken of a probe cut out of one of the cracked areas along the butt. It is to be noted in the upper or face macro, that the cracks seem to originate in the reinforcement bead adjacent to the base metal and progress inward through one or more adjacent weld beads. The macro of the transverse section shows that the reinforcement beads adjacent to either side of the weld are laid on base metal beyond the scarf.

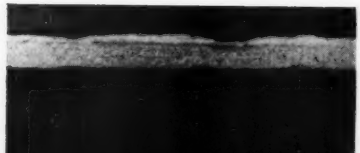
Figure 27 shows a photomicrograph of one of the cracks taken at 200 diameters. The etchant used was a 2 percent Nital. As shown, the crack is purely trans-crystalline indicating that it is a cold crack or one occurring at a relatively low temperature.

A hardness survey of a transverse section taken from the probe is shown in Figure 28. So far as can be reconstructed, the reinforcement layer was deposited in the numbered sequence shown in Figure 28. Beads 1, 2, 7 and 8 were deposited on the plate surface adjacent to the butt scarfs. As would be expected, this produced approximately 30 percent dilution of the weld deposits resulting in the development of an air hardening alloy steel. This is indicated by the hardness survey plotted below the sketch. Bead 8, the last to be deposited, has a hardness of 575 Vickers Brinell and it is probable that bead 7 had the same hardness before being tempered by the deposition of bead 8. The same is apparently true for beads 1 and 2. It is believed that the cracking experienced in the reinforcement layer was directly caused by base metal dilution forming an air hardening alloy whose suppressed transformation temperature developed sufficiently high internal stresses to produce cold cracks. This base metal dilution can be appreciably reduced by restricting the width of the reinforcement layer to that just sufficient to fully cover the butt scarf and by depositing the side beads with a slight weaving technique so that base metal contamination is kept to a minimum.


SCHEMATIC SKETCH SHOWING SEALING LAYER
DIAMOND PYRAMID HARDNESS NUMBER
(VICKERS HARDNESS—10 Kg. LOAD)
APPROXIMATE LOCATION OF HARDNESS READINGS

HARDNESS SURVEY TRANSVERSE REINFORCEMENT LAYER
DOUBLE "V" GROOVE WELD

Figure 28



SURFACES OF FRACTURE
FILLET WELD DEPOSITED ON CLEAN GROUND PLATE SURFACES
MAGNIFICATION 1.5X

Figure 29

SURFACES OF FRACTURE
FILLET WELD DEPOSITED ON RUSTY PLATE SURFACES
MAGNIFICATION 1.5X

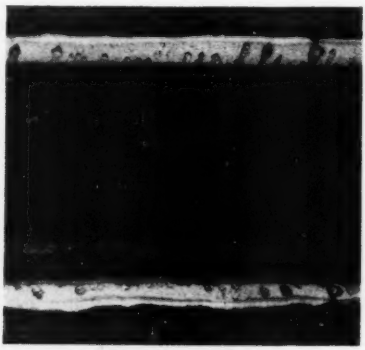
Figure 30

Figures 29 to 32 inclusive are a series of photographs dealing with fillet welds deposited on clean, rusty, galvanized or painted surfaces, with low-hydrogen AWS-E-7016 type electrodes. Figure 29 shows the surfaces of fracture of a fillet weld deposited on clean ground plate surfaces. The rather uniform texture of the fractured surfaces and the relative freedom of macro porosity are to be noted. However, as shown in Figures 30, 31 and 32, when fillet welds are deposited on rusty, galvanized or painted surfaces, the deposits when fractured show considerable "worm hole" type of macro porosity.

At first, it was believed that the water content of the paint pigment or that absorbed by the rust was the cause for this type of porosity. However, it now appears that the oxide rather than the water content of the pigment and rust may be causing this difficulty. Though it has not been fully established, preliminary work appears to indicate

SURFACES OF FRACTURE
FILLET WELD DEPOSITED ON GALVANIZED PLATE SURFACES
MAGNIFICATION 1.5X

Figure 31



SURFACES OF FRACTURE
FILLET WELD DEPOSITED ON PAINTED PLATE SURFACES
MAGNIFICATION 1.5X

Figure 32

that a reduction of the oxides takes place in the metallic arc and the released oxygen, in combined form, causes the "worm hole" porosity. However, the investigation of this phase of the work is continuing in hope of definitely establishing the cause and necessary means of prevention to insure against a recurrence in future work. In the meantime it appears necessary that all surfaces in way of fillet welds be cleaned prior to the deposition of such welds.

In concluding, it is believed that by proper use and observation of the precautions presented herein, the Ni-Mo-V type of electrode will fulfill the requirements necessary for the satisfactory electric arc welding of special treatment steel. The usability characteristics of the electrode do not differ materially from those of the AWS-E-7015 or 7016 class and the strength of the weld deposit appears to be sufficient to develop joint efficiencies of 100 percent in special treatment steel. In view of the excellent reception it has been given by various production activities, it is the opinion that the use of the nickel-molybdenum-vanadium electrode will have wide preference over the Navy Grade IV (25-20) austenitic type, not only by design agencies, but, also by construction yards, in the welding of special treatment and other low alloy, air hardening steels.

ACKNOWLEDGMENT

The authors wish to acknowledge the support and assistance given by the Navy-N.E.M.A. Electrode Development Committee; Comdr. R. H. Lambert, BUSHIPS; Messrs. C. A. Loomis, BUSHIPS; R. D. Thomas, Jr., Arcos Corporation; E. B. Lutes, Arcrods Corporation; D. C. Smith, Harnischfeger Corporation; L. K. Stringham, Lincoln Electric Company; D. E. Helm, McKay Company; D. L. Mathias, Metal and Thermit Corporation; and, G. E. Claussen, Reid-Avery Company. The authors are also grateful to Commander W. E. Pittman and Mr. J. E. McCambridge of the Industrial Test Laboratory for their encouragement in the preparation of the manuscript.

Official Department of Defense Photo

LCVP's after loading with troops are ready for invasion during recent exercise.

NAVAL USES FOR GLUED LAMINATED TIMBER

PETER W. MOURER, JR.

THE AUTHOR

Non-graduate USNA Class of 1926, Registered professional engineer States of Washington and North Dakota, author of technical articles ranging from refrigeration of mass concrete to development of strategic mineral resources, listed in "Who's Who On The Pacific Coast," division engineer for Timber Structures, Inc., Portland, Oregon, and partner in firm of consulting engineers, Seattle, Washington and Anchorage, Alaska.

Even prior to Unification, the older salts were prone to lament the passing of wooden ships and iron men. Therefore, this shall be confined strictly to the former—and other non-animate uses. For, despite recent contrary contentions, the Navy is still using forest products, and that rate of use promises to increase as the field of application for glued laminated timber becomes more widely realized among those responsible for the adaptions of available materials which have brought naval engineering the eminence it so rightly enjoys in the various fields of technical progress.

What is Glued Laminated Timber? I quote a considered definition from the literature of one of its leading manufacturers for marine use: "Laminated timbers are built up from multiple thicknesses of lumber, bonded together with synthetic resin adhesives of phenolic type to form an integral whole. Elimination or control of defects during fabrication assures a finished product of known and uniform strength free of hidden weaknesses and tailor-made to fit the requirements of a particular boat or design."

Need for one of the advantages that glued laminated timber affords was felt by the Service nearly a century ago, when commanders of naval vessels voyaging to the then remote Pacific Northwest region were charged with locating and securing for future naval use "stands of timber suitable for employment in the fabrication of masts and spars." It is also recorded that the freshly topped fir that served in place as flagpole for dedication of the fledgling naval station at Bremerton was later used to remast a naval craft neglecting to shorten sail off Tatoosh. Despite such precautions and husbandry, which at one time included an instruction to surveyors of the General Land Office that they be careful to record in their notes "exact locations and extent of spar timber," dimension timber is now a very

LCVP's after loading with troops are ready for invasion during recent exercise.

scarce item in solid timber. It is doubtful if the Navy of a century ago would be able to meet its needs from the stands of solid timber now available. Happily, since the advent of glued laminated construction, we are able to furnish from our depleting forests, "shop grown" timber superior in every way to solid timbers. These are available for aprons, deck beams, binding strakes, booms, capping, deadwood, frames, horn or counter timbers, keels, keelsons, knees, mast stepping, risers, shaft logs, side fenders, stems, stern-posts, strakes, stringers, towposts, transoms, decking, bulwark stringers, ladder rails, gun platforms, gangplanks, and a multitude of other applications of which our seagoing forbears never dreamed. These "shop grown" timbers are, contrary to general belief, not only stronger but far more durable than solid wood.

The significant increases in strength and durability found in exhaustive testing of glued laminated timbers, by the Forest Products Laboratory and other responsible and unbiased agencies, is directly traceable to close and selective stress-grading, alternation and staggering of grain slopes for maximum stress dispersal, culling of laminae showing knots or other strength-affecting defects, and development of adhesives stronger than the with-grain shearing strength of the wood. Considerable advances in the chemistry of glues and synthetic resin adhesives have been accomplished since its publication, but basic information on use-properties of over 100 compounds then available is contained in Reference 1. Application and curing methods have been revolutionized since publication of that comprehensive and valuable work by the development of high-frequency electrical apparatus which secures almost instant setting of glue lines. Values given in Chapter 7 of the subject reference for increases in design stresses are extremely conservative. Present design practice is concisely set forth in Reference 2, though it should be noted that basic stress values given therein are based upon static loadings, and should be modified for application to any dynamic or impact uses as commonly encountered in maritime work. Particular consideration should be given connections and fastenings of metal since maximum stresses usually occur at such points of transferal, and it should be borne in mind that impact stress will be sustained almost wholly by the least resilient material with consequent overstress thereof.

The uses mentioned above are but a fraction of the possible adaptions to naval construction of this versatile and easily procured material. While eclipsed by the lighter metals as basic structural material for aircraft; proof of its utility for such application was demonstrated forcibly by Howard Hughes in successfully flying his mammoth plywood seaplane. Present dimensional trends for larger aircraft indicate that the structural economy of the glued laminated parabolic arch may well be further applied in hangar construction. Cost, as well as weight, is appreciably less than steel. Strength-weight ratio of any structural material should be an important evaluation factor, particularly when considered from the standpoint of naval air logistics. Glued laminated members, prefabricated to exact requirements, compare favorably with other materials to an extent not generally realized, having more than 10 percent of the strength of steel and 7½ percent of its weight. Arch spans of well over 200 feet are entirely practical and, since thrust rather than moment is usually governing for the parabolic form, fabrication in segments for ease of transport may be employed. This form of structure, particularly with a "stressed skin" sheathing, has the added advantage of having more evenly distributed stresses under wind pressure; an important consideration when designing for typhoon or "williwaw" areas. Simplicity of erection of prefabricated glulam elements

by unspecialized labor is also a definite advantage in remote areas. The erection difficulties encountered by Construction Battalions with light-gage steel frames and corrosion problems later encountered by maintenance forces are minimized in glulam construction.

Dredge spuds, range markers, spar buoys, boat booms and similar gear admits of economical fabrication with greater strength and durability through use of glued laminated timber. Prior difficulties with delamination have been solved with developments in adhesives which are entirely waterproof and applied on continuous glue lines bonded by passage through a high-frequency electrical field, which has replaced the bulky and time-consuming curing chambers formerly used. Informative reviews of allied uses for maritime purposes and data on experimentation under Navy exploratory contracts are ably and concisely reported by Verne Ketchum, a civil member of the Society, and C. D. Dosker in Reference 3, which also contains a basic explanation of the high-frequency gluing process by P. D. Zottu.

Douglas-fir is more extensively used for maritime purposes than any other wood and is the only material used in the major fabricating plants of the Pacific Coast. For those designers confronted with the problem of design in other American woods, Reference 4 will be found to contain full data on structural properties of practically all woods indigenous to the United States, together with details for simple and effective test methods which might be easily applied to specimens of foreign woods on which such data might be sought. Experiments at an eastern college of forestry have gone into the advisability of combining species in a single member with surprising results. The field for further experiments in both materials and technique is broad, varied, and certain of beneficial results.

The unprecedented expansion of naval technical activity during the past decade makes it imperative that all engineers examine their experience objectively for possible summations of information which, for the good of the Service, might well receive emphasis and review. That thought has guided the preparation of the foregoing and no claim for originality can be made. Whatever of that quality exists derives from study of the References cited and other of abundant sources. The aim has been only to outline beneficial and practical uses for an abundant and economical material with due emphasis on recent improvements in design and fabrication techniques. Basic design data for the uses set forth can be found in the References and other sources noted therein.

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Official Department of Defense Photo

The Viking Rocket, loaded with instruments for upper atmospheric research, was fired at White Sands, New Mexico, on 9 February 1950. This was the third Viking fired. When it appeared that the rocket was veering slightly off the range to the west its ascent was cut off from the ground. The maximum altitude attained was 51 miles. The Naval Research Laboratory reports that the telemetering was excellent and the upper atmosphere research was satisfactory.

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PART TWO

REPRINTED ARTICLES

The articles contained in this Part Two represent those which, in the opinion of the editorial staff, are the most valuable which have been gleaned from recent issues of contemporary publications.

This material was formerly published under the heading "NOTES". It has been given somewhat more prominence and made more readable to enhance its value. The editors' purpose here is to provide a convenient repository to readers of the JOURNAL, for articles originally published elsewhere which contribute to the purpose of the Society: to further the advancement of naval engineering.

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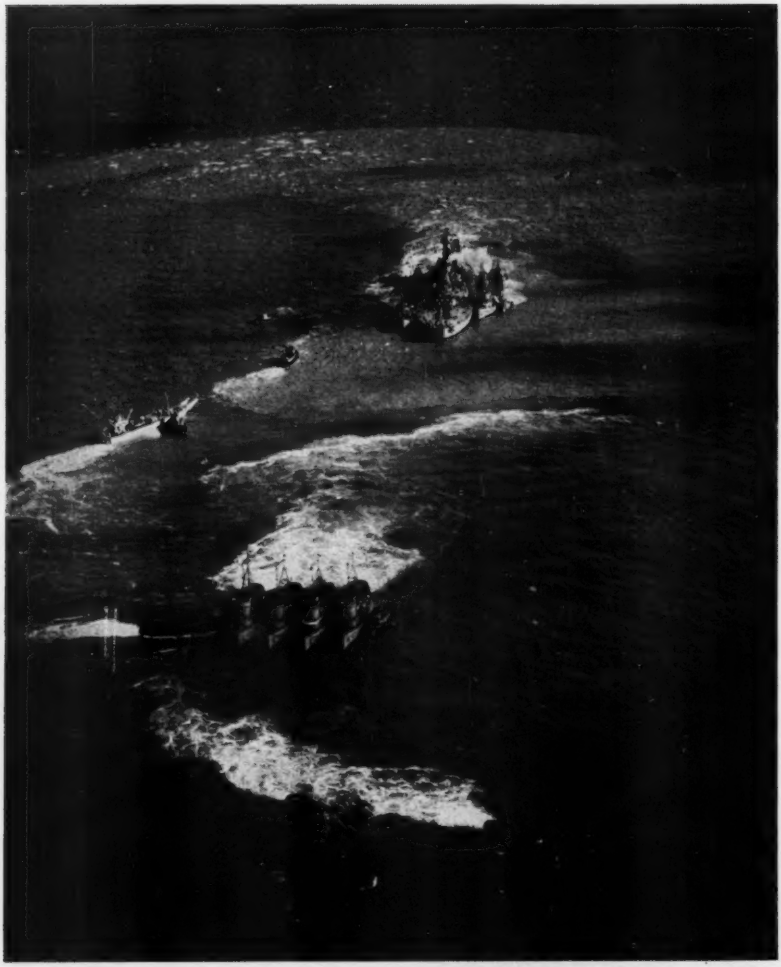

Official Department of Defense Photo

USS *Missouri* (stern view) which ran aground in Thimble Shoals in Hampton Roads, Virginia, is shown with fleet and harbor tugs alongside and aft during one of the four attempted "pull off" operations, which failed to move the 45,000-ton ship from the sand bar. After dredging alongside and astern toward the main channel, and aided by a northeast high tide, tugs and winches, the *Missouri* broke her bonds during the fifth try.

THE IMPORTANCE OF OUR MERCHANT FLEET

ACKNOWLEDGMENT

VICE ADMIRAL E. W. MILLS, USN (Ret.), President of Foster Wheeler Corp., past president of the Society, is the author of this article that appeared in the May, 1950 issue of the *Nautical Gazette*.

In every war American tactics have been to carry the fight to the enemy. From discussions which have taken place recently with respect to the American Merchant Marine, it would appear that this cardinal principle of carrying the fight to the enemy may be in danger of abandonment. Statements have been made that it is not necessary for this country to maintain an adequate Merchant Marine. The only reasons given to back up these statements are, first, that it costs us too much to build and operate ships and, second, in the event of another emergency, we can depend upon our Allies to furnish the ships for carrying our war supplies. Statements such as these are extremely fallacious, and they do not indicate a basic understanding of the factors which have made this country invincible in past military efforts. In addition to the lesson which we should have learned following World War I, the tragedy of Pearl Harbor served to emphasize most strongly the need of this country for combat and merchant ships. We literally rode to victory on the merchant ships that were produced in World War II after strong Allies gave our country a chance to mobilize its industrial potential and its capacity for building ships. We have no assurance that in the next emergency we will have strong Allies to hold back an enemy while we prepare ourselves; nor have we any assurance that our Allies will be nations who are strong in sea power. It would seem that with the lessons of two World Wars relatively fresh in our minds no writer should have to argue regarding the necessity of our having a strong Merchant Marine as a vital element in our military strategy and logistics; without which none of our Armed Services can be readily effective. That one reason alone should be sufficient to convince anyone in authority that this nation should take whatever action is necessary to insure that our Merchant Marine can be built up and operated under private ownership and management in peacetime. This can be done only by substantial help

from the Government in the form of subsidies; but it is not just, nor does it show a proper understanding of the problem, to condemn these subsidies simply because they involve Government assistance. Actually, the Merchant Marine Act, which provides for subsidies in the construction and operation of merchant vessels, might well be reduced to the simple word "parity," because the Act, through the so-called subsidy, only provides American ship builders and ship operators equality of opportunity with foreign-flag competitors.

Criticism which may be leveled at the subsidy principle is not valid because, in simple words, it is unthinkable that we should place ourselves in a position where in times of emergency, we shall have to depend upon other nations to furnish our carrying capacity. In addition to our not knowing whom our Allies may be in a new emergency, it should also be pointed out that in World War II there was no nation or combination of nations which could have furnished all the tonnage we found was needed. A powerful nation like ours cannot afford to depend upon our Allies for sea transport, because if we do, we are likely to find ourselves in a position where we cannot develop and deliver the tremendous industrial potential which this country possesses. We must be able to implement our own military and naval efforts promptly, when the emergency arises, and not be forced to depend on some combination of Allies to furnish help in the way of ships, if we are going to continue our time-honored principle of carrying the war to our enemy and fight it to a successful conclusion in his territory. I frankly cannot understand the attitude of any agency in the Government which directly or indirectly advocates that a Merchant Marine under the United States flag is unnecessary to our national security. Actually, the survival of American shipbuilding and the maintenance of a strong and adequate Merchant Marine is an essential element in our national security which is second to no other.

The lesson to be learned from our two world wars should be abundantly clear, that if this nation ever again becomes embroiled in world-wide conflict, powerful military and air forces alone will not be sufficient. If we wish to continue to pick the location of the fight, it is absolutely necessary that we maintain and operate a great armada of cargo and transport ships with the shipbuilding facilities for creating more if an international crisis should so demand. As a nation, we fervently hope for continued peace, but we must not be so naive as to assume that peace can be assured by hoping for it. We must take a much more realistic view and determine to keep ourselves strong and ready for any eventuality if we should again become the victim of an aggressor nation. If we keep ouselves at the proper strength, we will discourage any ideas of conquest that an aggressor might have, because no nation enters into a war when she thinks she does not have more than an even chance of winning.

The Navy amply proved in World War II that it was both daring and courageous. Even when we were outnumbered in the Pacific, we struck some of our most vicious blows at the Jap Navy, which were instrumental in turning the tide of war and in opening up the paths to ultimate victory. Our new combat ships had greater speeds and longer cruising radii than the Japs believed possible. They could not understand our capacity to remain in continuous active service over such long periods of time. They could not match our performance in either operation or building of ships, and it must be recorded to the everlasting credit of the shipbuilding industry that it produced a Navy and a Merchant Fleet which gave the Japs a definite impression that our sea power was many times stronger than it was in reality. Every invasion,

every victory, stemmed from ships, and from what these ships were able to carry, and from what these ships were able to do in destroying the enemy control of the sea.

We do not know definitely what form future wars may take. With the advent of the atomic age and the development of guided missiles, we have to speculate somewhat as to the types of weapons that may be used against us in the future. But we can be sure of one point: we know that should another war come, whether it be fought with guided missiles or atomic weapons, or with some new air-borne device not yet conceived, we will need sea-going vessels—and lots of them—if we are going to fight in the territory of the enemy. During the latter part of World War II, much of our fighting was done by far-reaching Navy ships and with long-range bombers. This situation, however, did not reduce the need for cargo and transport vessels. On the contrary, as we threw more and more of our strength at Germany and Japan, the demand for carrying tonnage greatly increased. Some scientists like to infer that the next war may be a "push-button" affair, which conceivably might be concluded in thirty minutes, *if* we have the right assortment of weapons. Should this fantastic development be realized, the conquest would still have to be followed up with troops and supplies, and we cannot foresee an age when sea-going vessels can be supplanted by more reliable and economical methods of transporting vast quantities of supplies across the oceans of the world.

Expanded fighting fronts, whether they be air bases, naval bases, or military bases, when they are thousands of miles from our shores, automatically set up a demand for more and more transport and cargo tonnage. Every fighting man sent overseas requires an initial shipment of about five tons of organization equipment, and he then requires approximately an additional ton per month to maintain him abroad. When you have over five million troops overseas, the logistic demands are tremendous. In the past, given time, our American shipyards met the challenge, but we must be sure that in the next emergency we either have these ships in being, backed by the facilities for rapid expansion; or we have the assurance that ample time will again be available. The latter alternative we cannot guarantee; so it would appear that the course of wisdom indicates that we have the ships either in operation or in reserve.

The conclusion, then, is unavoidable, because no matter what types of weapons we adopt, or what security program this nation follows, we cannot neglect our Merchant Marine. Our transport and cargo ships are just as much an integral part of our national security as any of our Armed Forces. Our private shipbuilding industry must be kept healthy and active during the peacetime years if we are to expect it to be able to meet the unusual demands of another national emergency. The United States has taken its place as a world maritime power and from the standpoint of national security alone, we cannot afford to relinquish that position. It is essential that we take steps to insure that merchant vessels, flying the Stars and Stripes, continue to navigate the oceans of the world and to participate in international trade. It is a must that we put the Merchant Marine on a good sound operating basis with inducements to the ship operators to build new ships more suited to the needs of commerce and war than some of the present tonnage. If this result could be achieved, it would assure the continuance of a shipbuilding and a ship repair industry which is a vital part of an adequate and successful Merchant Marine. It is inconceivable that the shipbuilding and ship repairing industry should be allowed to wither and dissipate its skilled men to other industries.

because such a result would face us with the desperate necessity for again reorganizing in the event of emergency. When a nation ceases to be interested in being a maritime power, it takes its first step in destroying a vital element in sea power, and I believe that history has amply demonstrated time and again that no world power has been able to continue without a proper appreciation of the importance of sea power. It has been truly stated that military, naval, and air power are all seriously circumscribed without a strong supporting Merchant Marine, to say nothing of the fact that world trade cannot be developed and maintained without it.

We must support a strong Merchant Marine because it is a vital element in our national security; and the subsidies necessary to insure parity with foreign competition for our American ship builders and ship operators mean money well spent.

HYDRAULIC SEALS

ACKNOWLEDGMENT

The Institution of Mechanical Engineers, Proceedings for 1949 contained this article on Hydraulic Seals. It is divided into two parts: Rotary and Oscillating Seals by Mr. T. E. BEACHAM, and Reciprocating Seals by Mr. F. H. TOWLER. It deals with hydraulic machinery in which oil is the working medium. It does not include hydraulic joints but rather the seal between surfaces having relative movement.

ROTARY AND OSCILLATING SEALS

BY T. E. BEACHAM, B.SC. (ENG.), M.I.MECH.E.

Introduction. Rotary oil seals with the packings now available do not present much practical difficulty at slow and intermittent speeds even where high pressures are involved, but at high speeds wear and heat due to friction become important factors.

Customary types of stuffing boxes can be manually adjusted when excessive leakage occurs, but many types of non-adjustable or automatically adjustable shaft seal are now used; there is little information available concerning the life of the latter which may be expected before the leakage becomes such that replacement is necessary. Standardized life tests under laboratory conditions would be of great value not only in comparing one type of seal with another but in estimating the probable life under practical working conditions.

The life will in many cases be affected by vibration due to shaft eccentricity, shaft deflection and wear of the journal bearings, and by wear caused by the abrasiveness of the dirt in the liquid. It should, however, be possible to allow for these factors and use standard life test results with an appropriate factor of safety.

To obtain reliable results for any particular type of seal under given conditions it would be necessary to test a number of seals and plot a curve as shown in Fig. 1. The minimum life expectancy of any proportion of the seals

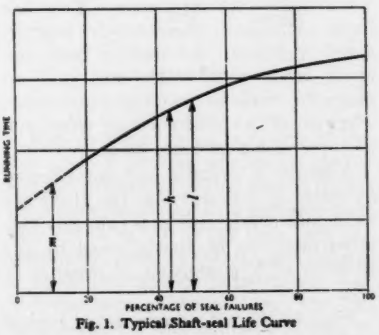

Fig. 1. Typical Shaft-seal Life Curve

An alphabetical list of references and bibliography is given in the Appendix.

can be taken from the curve; for example, m would be the minimum life expectancy for 90 percent of the seals and l for 50 percent of the seals. If the curve is plotted up to the 100 percent failure point, then the mean height h of the whole curve will give the true average life.

There is no generally accepted standard as to the amount of gland leakage that can be tolerated. In many cases a considerable leakage may not affect the operation of the plant, but with a moderate rate of loss, oil is apt to form an unsightly pool and a limit has to be imposed by considerations of cleanliness. In other cases, however, where a limited amount of liquid is involved—for example, in hydraulic couplings—it is important to have the minimum amount of leakage.

The standard of 10 cu. cm. per hour leakage laid down for British aircraft pumps is reasonable for small high-speed pumps where the practical limitation is one of cleanliness. On the other hand, United States Navy specifications, for some purposes, give a gland leakage of one drop in fifteen minutes, which is equivalent to approximately ¼ cu. cm. per hour.

Soft Packings. Soft packings of organic material such as hemp, cotton, and flax are often less effective with oil than with water, as oil tends to dissolve the fatty substances with which these packings are commonly impregnated. Moreover, the fibers of such material swell with water and tend to make the mass of material impervious, whereas oil has a dehydrating effect and the reverse action takes place.

Asbestos is organic and not affected by water in this way, but the fibers are weak and brittle and as a rule this material needs to be strengthened by the addition of a proportion of organic fibers.

Fig. 4*a* is not shown as a typical stuffing box but is of interest on account of the hardened bush used and the limited space available. The gland was fitted between two sections of a duplex gear pump arranged to operate two aircraft gun turrets. The pressure difference varied from zero to a maximum of 15 lb. per sq. in. The shaft bush was of nitrided steel running at a maximum surface speed of 800 ft. per min. A moulded asbestos packing was used originally but it disintegrated after a few hours running, and was replaced by asbestos plaited round a white metal wire, wound into a spiral and compressed to the stuffing-box dimensions; this gave satisfactory results. The packing became progressively looser with use, apparently due to fibers being broken up by contact with the shaft and then washed out by the liquid, but it remained serviceable without adjustment for the periods between pump overhauls, until the gland could be tightened up. Wear on the surface of the shaft bush was negligible. Hardened shafts are sometimes used in industrial pumps to prevent wear by the gland packing, and in some cases shaft sleeves of Stellite are used.

Leather Seals. Leathers, even of the chrome tanned variety, deteriorate with time when they are in contact with mineral oil, especially when hot. A certain amount of leakage through the pores of the leather takes place under pressure but the material will prevent ingress of air under slight vacuum, apparently by a capillary effect. Leather seals cause less shaft wear than any other type of seal.

Fig. 4*c* shows a spring-loaded hat leather used as the outer seal in an engine-driven aircraft pump; its function is to act as a seal between the engine crankcase and the atmosphere. The initial contact is provided by an axial spring which presses the leather to the shaft by means of a conical washer; in other types of seal, garter springs are often used.

Rubber and Synthetic Rubber Seals. Minimum leakage can only be achieved

with seals of impervious materials such as rubber, cork, metal and carbon. Rubber and synthetic rubber are the most effective materials for preventing seepage of oil through a moving joint, and for rotary seals having slow or intermittent movements rubber is the ideal material. Synthetic rubber is now usually employed with mineral oils to obviate their swelling and deteriorating effect on natural rubber. In the remainder of the paper the term "rubber" is used to include synthetic rubber.

Fig. 2 illustrates a number of seals using moulded rubber packings of various forms for slow and intermittent rotation such as occur with valve spindles.

Fig. 2*a* shows the S.E.A. ring which is made of several layers of fabric impregnated and bonded with rubber and moulded to shape. This packing was introduced nearly forty years ago, and

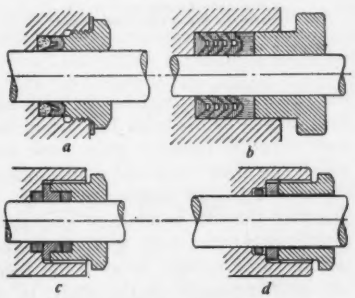

Fig. 2. Packings for Slow-speed Shafts
a S.E.A. ring. *b* Chevron packing. *c* Square-section ring. *d* O-section ring.

was the first effectively to prevent oil seepage through a moving joint. Two or more rings can be used in series, and normal methods of gland adjustment can be used; where one ring only is used it is desirable to use a turn of soft packing or a moulded ring of semi-circular section to provide initial contact between the lip and the shaft.

Fig. 2*b* shows the chevron type of packing; the two end chevrons are sometimes used without the intermediate ones.

There are many other forms of moulded packing, of solid rubber or impregnated fabric, which give satisfactory results.

Figs. 2*c* and 2*d* illustrate the use of square- and O-section rings respectively. There is no adjustment for wear except that provided by the initial compression of the rings themselves, and they are mainly used for occasional movements.

For high-speed shafts a rubber seal tends to prevent oil penetrating between the shaft and the rubber surface in contact with it, thus causing high coefficient of friction, with excessive heat and wear.

There is a striking difference between the friction of leather and rubber under these conditions. Experimental results which have so far been published have not given much information on the subject probably because they refer to low speeds and surfaces having a limited length in the direction of motion. The effect of temperature on the friction of rubber is also a matter on which more information is desired.

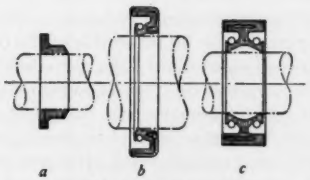

Fig. 3. Rubber Shaft-seals

Fig. 3*a* shows an early design of rubber seal, of similar section to the hat leather previously mentioned. The seal gave good results at an intermittent surface speed of 50 ft. per min., but at a continuous speed of 400 ft. per min. with a small interference between the shaft and the bore of the packing, the local heat developed was sufficient to blue the shaft which soon became heavily scored. Fig. 4*a* shows a gland separating pumps,

and Fig. 4*c* an engine crankcase seal (hat leather).

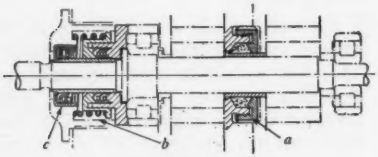

Fig. 4. Aircraft Pump Seals

The spring-loaded chevron packing (Fig. 4*b*) was used successfully as a shaft seal on many aircraft pumps. The chevrons were originally made from fabric impregnated with synthetic rubber but these were insufficiently pliable and were subsequently replaced by chevrons of solid synthetic rubber. The interference fit necessary to give satisfactory results was extremely critical and there was a certain amount of difficulty in moulding the chevrons to the required limits. The shaft was provided with a sleeve of nitrided steel which was very hard, to resist wear, and did not soften at the surface when subjected to occasional overheating, in the same way as case-hardened shafts.

One solution of the heating difficulty appears to be to make the contact area between rubber and shaft extremely narrow; seals of the type shown in Fig. 3*b* are now in common use for small pressures. This type of seal is simple, easily fitted, and occupies but little end room; the interference fit between seal and shaft is much less critical than with the chevrons, and the garter spring is used to provide some initial contact pressure in addition to that due to the fit. For many applications this seal gives good results with soft shafts. In other cases, however, particularly with thin oil containing abrasive matter, a certain amount of shaft wear occurs and it is advisable to use renewable, hardened shaft bushes.

Fig. 3*c* shows a compact form of this seal with twin lips to form a seal against both pressure and vacuum. This was used on many of the later aircraft pumps in place of the chevron packing.

The relative advantage of hard and soft shafts, the correct hardness for the rubber, the optimum surface finish of the shaft, the life expectancy, and its variation with the pressure on the seal, are matters on which information is required.

Cork Seals. Fig. 5*a* shows a spring-loaded stuffing box filled with virgin cork, used for the shaft of a petrol pump.

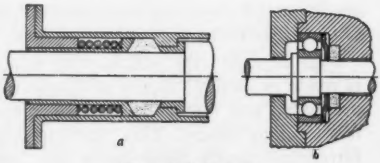

Fig. 5. Cork Shaft-seals
a Spring-loaded stuffing box.
b Cork ring used in Vickers' vane pumps.

For this purpose cork seals have now been largely superseded by carbon face seals. Fig. 5*b* shows a plain cork bush which has been very successful against low pressures, for certain oil pumps. In this case the seal is made of granulated cork impregnated with synthetic rubber and the bore ground to give the requisite interference fit on the shaft. In the pump for which this seal is most used, the shaft is not subject to any journal load but is used only to transmit torque.

Metal-to-metal and Metal-to-carbon Seals. Gronau (1936), gives details of a number of types of metal packing which can be used in stuffing boxes and adjusted in the ordinary way. The recent tendency is, however, to use constructions for metal and other rigid materials in which their rigidity is an advantage and not a difficulty to be overcome.

Journal Seals. A metal-to-metal seal, consisting of a bronze sleeve tapering to an edge with the bearing area on the

shaft reduced almost to a line, has given promising results but so far as the author is aware has not been used much in practice. Like the rubber seal in Fig. 3*b*, the bronze sleeve has a slight interference fit on the shaft.

Packings composed of split rings of metal or carbon surrounding the shaft and pressed against it by garter springs (Fig. 6*a*), although used for other purposes, do not seem to be favored for shaft seals for oil.

Fig. 6*b* shows a seal composed of two split carbon or plastic rings embedded in a rubber matrix. This is a recent development which combines to some extent the advantages of both rubber and metal-to-metal seals, as it avoids the heating difficulties of the former and yet is simple and easy to fit. The section of the ring is designed so that pressure from the interference fit is uniformly distributed, and the gaps in the rings are diametrically opposite each other; the rubber is moulded around the rings and prevents the leakage of liquid passing from one gap to the other. At

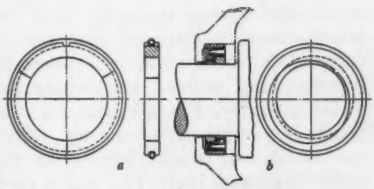

Fig. 6. Metal and Carbon Journal Seals
a Split carbon ring. *b* "Jaru" shaft seal.

zero pressure the rings and shaft are in contact by virtue of the small interference between the shaft diameter and the bore of the rings. Any liquid pressure on the seal is transmitted through the rubber to increase the pressure exerted on the shaft, and thus the rings are enabled to maintain contact and prevent leakage.

The designer stresses the advantage of a highly polished surface for the shaft and advocates the use of a shaft sleeve if it is impracticable to superfinish the shaft itself.

Face Seals. Face seals probably offer the greatest possibilities for use with high-speed shafts. They are suitable for particularly difficult sealing problems, such as occur at the crankshafts of refrigerating compressors where it is necessary to prevent the escape of noxious gases under pressure, and the shaft seals of fluid couplings, where a limited amount of paraffin or light oil has to be retained without undue loss.

The rubbing speed is higher with face seals than with journal seals, but, with the former, both wearing faces are independent of the shaft and are formed on minor parts which can be readily renewed, although, of course, in common with most of the newer types of shaft seal, the replacement parts have to be threaded over the end of the shaft. It is claimed that face seals can be designed so that the sealing is unaffected by substantial misalignment of the opposing faces, or by a limited amount of shaft eccentricity, and that they can be arranged so that the faces are not damaged by abrasive particles in the liquid, and that they can seal pressures up to some 300 lb. per sq. in.

There has been considerable detail development in the construction of seals of this type; Fig. 7 shows only a few of the various types which are in use.

One face usually consists of a bronze or carbon ring and the other face may be of hardened steel, mechanite, cast iron or bronze, Carbon rings are more suitable for liquids of low lubricating value, such as water, petrol, paraffin and light oils. For heavier oils bronze rings are favored, but many varieties of carbon and carbon-metal mixtures are now available and their use for seals is extending; the advantage claimed for carbon is that it develops a hard polished surface with a low rate of wear.

One of the sealing faces must be able to move axially; Fig. 7*a* shows a simple

type in which a bronze ring A is attached to the casing, the mating element B being of hardened steel. The latter can slide along the shaft, the initial contact being made by the spring C. To prevent leakage between B and the shaft a rubber seal D is used. This and similar types are largely used for small refrigerators and industrial oil pumps. Twin seals acting in opposite directions are often used to deal with both pressure and vacuum but when two seals are used in this way the cooling and lubricating of the outer one has to be considered. The friction between the ring D and the shaft will increase or diminish the effective force of the spring and in some cases this friction will impair the efficiency of the seal. In the aircraft pump seal (Fig. *7b*), an O-section ring is used which offers less friction than that of rectangular-section rings; the element B is driven by dogs while the fixed ring A is not made a tight fit in the casing but is bedded on to another O-section ring.

Another construction which has been much used in aircraft pumps is shown in Fig. *7c* where the drive is taken through an Oldham coupling. Faces are formed on two sections of the coupling and these are forced against the fixed rings A_1, A_2, by the spring C. This use of an Oldham coupling not only eliminates the objectionable axial friction previously mentioned but also provides a drive for the rotating faces. The rotating faces were of nitrided steel and the fixed rings were made of a carbon-metal mixture, and the arrangement formed the double seal of an engine-driven fuel pump running at approximately engine speed; the seal was effective with aviation petrol at pressures up to 20 lb. per sq. in.

In many applications the adjustable face is mounted on metal bellows which provide frictionless axial movement and transmit the torque due to the seal friction; the bellows make the necessary joint between the adjustable face and the shaft or casing.

Fig. *7d* shows an arrangement in common use for many types of refrigerating compressors; the preferred mating surfaces, when ample oil is available for lubrication, are nitrided steel and bronze.

Fig. *7e* shows a variation of the previous arrangement in which a carbon seal ring is free to float between two mating faces. This construction is used for fluid couplings and sometimes both mating faces are mounted on bellows. It contains no rubber and this is an advantage at high temperatures (Ruhl 1948).

Synthetic rubber is an alternative material for the bellows, and its use should obviate difficulties due to crystallization of the metal under vibration. In Fig. *7f* the rubber bellows are not used to transmit the torque to the adjustable face which is driven by the splines S. The mating face is resiliently mounted on a rubber ring E.

Figs. *7h* and *j* show the adjustable rings mounted on rubber and metal diaphragms respectively. These arrangements take up less end room than the others shown, and are an advantage in many cases where a long seal cannot be accommodated without lengthening the whole assembly, Fig. *7k* shows a seal which is exceptionally compact, where one mating surface is formed on the metal diaphragm itself. This seal is used on traction types of hydraulic coupling where the relative rotation is only that due to the slip of the coupling.

The softer materials (rubber, cork, etc.) tend to flow under pressure and behave as imperfect fluids, so that the internal pressure normal to the surface is equal to or greater than the liquid pressure trying to escape. Carbon and metal surfaces are substantially rigid under ordinary conditions and it is possible for the pressure between the faces to be less than the pressure of the liquid being sealed. With metal bellows it is

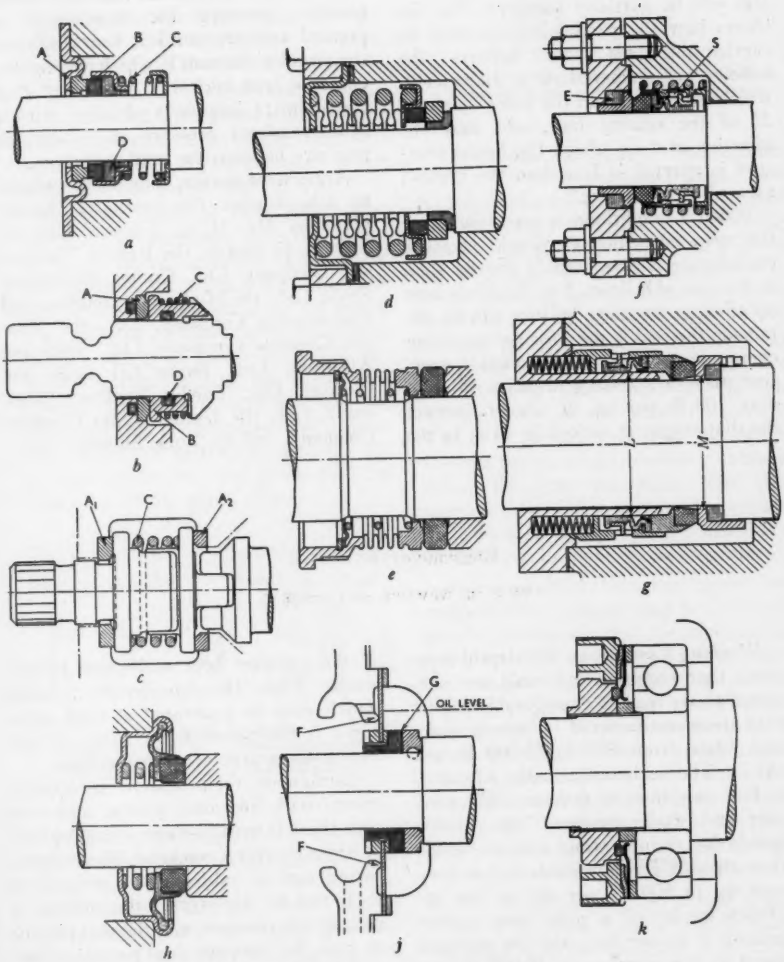

Fig. 7. Metal and Carbon Face Seals

a With rubber shaft joint.
b With O-section rubber shaft joint.
c With Oldham coupling drive.
d Metal-bellows type.
e Metal bellows with floating carbon ring.
f Rubber-bellows type.
g Rubber-bellows, pressure-balanced type.
h Rubber-diaphragm type.
j York "Balanseal" metal-diaphragm type, pressure balanced.
k Special metal-diaphragm type (for hydraulic coupling).

usual to arrange the diameter and anchorage of the bellows so that any pressure that the liquid can exert on the seal will be partially balanced. Fig. 7*g* shows how the partial balance may be carried out with rubber bellows; the bellows are anchored on a diameter *L* which is greater than the inner diameter *M* of the sealing face, and thus the effective area on which the liquid pressure is exerted is less than the contact area of the seal face.

When a diaphragm is used, the effective area on which the liquid pressure can operate is considerably greater than in the case of bellows. Fig. 7*j* shows how an effective pressure balance can be obtained; the seal is used for refrigerating compressors where the crankcase pressure may vary from a negative value to over 100 lb. per sq. in. Under vacuum the diaphragm is sucked in (i.e., to the right in Fig. 7*j*) and the negative pressure over the whole effective area may be sustained by the seal face G. Under positive pressure the diaphragm is pressed outward until it bears against the circular fulcrum F which divides the effective area and is so positioned that approximate balance is obtained with a small resultant pressure increasing the pressure between the mating faces.

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Reciprocating Seals

BY F. H. TOWLER, M.I.MECH.E.

Working Conditions. The liquid pressures that reciprocating seals are subjected to are usually considerably higher than those encountered by rotary seals, and range from 500 to 10,000 lb. per sq. in. The seals are usually subjected to two conditions of pressure: full pressure and slack pressure. The rubbing speeds for reciprocating seals are lower than those for rotary seals: at a pressure up to 500 lb. per sq. in. the approach stroke of a press ram seldom exceeds 1 ft. per sec., and the pressing speed at pressures up to 10,000 lb. per sq. in. seldom exceeds 3 in. per sec., and is often less than ½ in. per sec.

The "swept area" of a reciprocating seal is usually very large: for a gland seal it is equal to the circumference of the ram multiplied by the stroke, and for a piston seal it is the circumference of the cylinder bore multiplied by the stroke. Thus, the considerable friction which may be generated at high pressures is dissipated in the form of heat over a large area of wetted surface.

Continuous reciprocation is unusual except with hydraulic pumps, and even then there is usually some period of rest, at night or over a weekend. The reciprocating seal is subjected to reversal at each end of the stroke, the motion is usually intermittent, and the seal may be at rest for considerable periods either under full or slack pressure.

The viscosity, lubricity and chemical reactions of the liquid to be sealed—which may be oil, water or acid, or a combination of the three—also have to be considered in the design of reciprocating seals. In recent years the author's experience has been largely confined to

the oil hydraulic system, and therefore much of the paper is devoted to oil seals. It can, however, be shown (Gronau 1936) that the behavior of any seal which is oil lubricated is very similar, if not identical, to that of an oil seal.

Types of Seal. Reciprocating seals may be divided into: gland seals for piston rods or press rams; piston seals, and packingless seals depending upon a lap fit between the piston and the bore.

Hydraulic Packings and Sealing Materials. Soft packings, such as hemp, which are packed into the stuffing box and squeezed into contact with the sealing surfaces by tightening up the gland ring, are very common for water pumps but are not recommended for oil hydraulic systems because the packing is liable to disintegrate and get into the fine clearances of the pumps and valves. This is a crude method of sealing; the friction is excessive, and varies with the tightening of the gland ring.

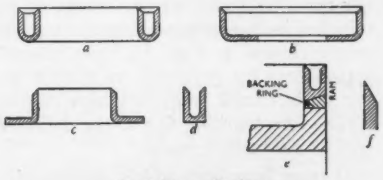

Fig. 8. Types of Packing

a U-section packing. *d* Square-bottom packing.
b Cup-shape packing. *e* Oblique-angle packing.
c Hat-shape packing. *f* Tapered-lip packing.

Automatic packings, of the U-leather, "hat"-leather and "cup"-leather type (Fig. 8), made of oak-tanned and chrome-tanned leathers have proved highly satisfactory for water hydraulic presses, and have been in existence with little change of form ever since the hydraulic press was invented in 1795. Chrome-tanned leathers are sometimes used with oil, but oak-tanned leathers are not suitable for use with mineral oil which tends to dissolve the natural fats in the leather and make it brittle, particularly if the oil gets warm.

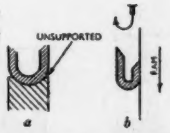

Fig. 9. Disadvantages of Round-bottom, U-shape Packings
a Feather-edge support. *b* "Roll" of round-bottom packing.

Fabricated packings, impregnated with synthetic or natural rubber, have largely replaced leathers for use with oil; these were first similar to the original leather packings shown in Fig. 8*a*, but as it was difficult to support the U-shape, see Fig. 9*a*, and there was also a tendency to roll, as illustrated in Fig. 9*b*, the present practice, especially for synthetic rubber is to use the square bottom packing shown in Fig. 8*d*. The oblique angle of the packing shown in Fig. 8*c* is advantageous, as it tends to prevent the packing from being extruded down the ram clearance. The thin, tapered lip of these packings, shown in Fig. 8*f*, is likely to remain because it is so effective in holding slack pressure.

It is essential that the curves and angles of the packing must be accurately produced in order that the packing may be effective; and it is difficult to ensure this unless both the packing and the backing ring are supplied by the packing manufacturer. The square-bottom, U-packing is the simplest to manufacture and fit, and is almost universally available. The same applies to square-bottom 'hat" and "cup" packings, except that they are not liable to roll because they are tightly nipped on the flange.

All automatic packings should be provided with some type of spring-loaded spreader ring to hold the lips of the packings in contact with the sealing surface, and prevent leakage past the packing when there is little or no pressure in the cylinder, and to prevent excessive leakage when full pressure is applied, until the lips have been opened out by the difference of pressure caused by the leakage. The spreader ring shown in

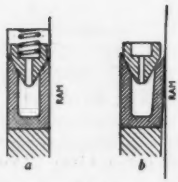

Fig. 10. Spring-loaded Spreader Ring
a Correct position. *b* Incorrect position.

Fig. 10*a* is not ideal because it is rigidly cylindrical, and may be effective on only one leg of the packing as indicated in Fig. 10*b;* a spring wedge of the form shown in Fig. 11 would be more effective. Slack pressure is usually more difficult to retain than full pressure, and considerable leakage may occur during periods when a press is not in operation.

The friction of automatic packings, if properly fitted, is very low, and as friction usually produces wear, this type of packing should have a long working life.

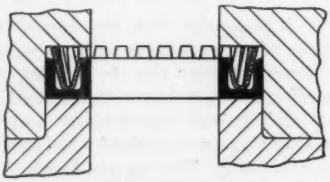

Fig. 11. Wedge-spring Spreader Ring

Chevron-type packings are a compromise between the soft packing and the automatic packing. The rings can be broken and wrapped round the ram instead of being threaded over it, but they have the disadvantage that they cause greater friction and wear; this is partially compensated as there is more than one packing ring to wear; but the rings also tend to wedge with high pressure, causing a great increase in friction.

Cast-iron piston rings have been used successfully in America for pressures of 2000–3000 lb. per sq. in., and in England for pressures of 6000 lb. per sq. in., but metal ring packings are only suitable for use with oil. The friction may be greater than with automatic packings, but as the wearing surfaces are metal, the wear is almost negligible. Three to six rings are generally used, although the question arises as to how many rings are really effective, and the nature of the pressure built-up behind each ring.

A combination of synthetic rubber and metal seems to have great possibilities: many rubber bonded rings are now being put on the market.

Carbon piston rings have been used on air compressors, and it is possible that they may have some application as hydraulic seals.

Synthetic rubber piston rings and O-rings may be classed as automatic packings. The free outer diameter of the ring is greater than the bore of the cylinder so that, when fitted in the cylinder, the ring is in sufficient compression to hold it in contact with the cylinder bore. On small bore cylinders, up to 6 inches in diameter, rectangular section rings, shown in Fig. 12, have proved satisfactory for pressures up to 6000 lb. per sq. in.; these rings may have two or three years working life if the cylinder bore is internally ground.

O-rings have been widely used on aircraft hydraulics in America because of their ease of assembly. They are suitable for pressures up to 1500 lb. per sq. in.,

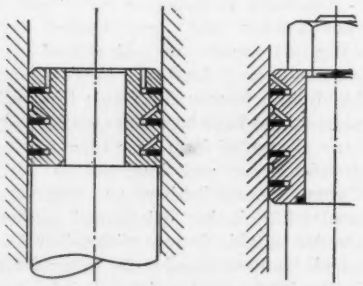

Fig. 12. Synthetic-rubber, Rectangular-section, Piston Rings.

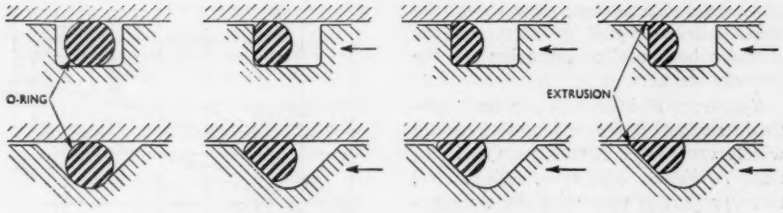

Fig. 14. The Twisting and Extrusion of O-rings with Increase of Pressure

and with the addition of backing rings, shown in Fig. 13, Hayman (1945) said

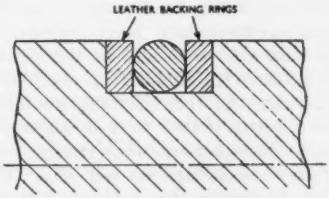

Fig. 13. Leather Backing Rings

that they were suitable for pressures up to 3000 lb. per sq. in. Fig. 14 shows how O-rings tend to wear flat, and to twist; the same feature occurs with small square-section rings, which will make a complete turn in their groove, unless they are tightly nipped (White and Denny 1947). It is for this reason that a rectangular ring is preferred.

The Mechanism of the Automatic Seal. White and Denny (1947) have demonstrated the mechanism of the automatic seal under static conditions; Fig. 15 shows the stress distribution across the faces of a seal. The life of a packing largely depends upon the surface finish of the ram, rod, or cylinder bore, and upon the resistance to extrusion, and the latter depends upon the clearance between the surfaces and the hardness of the material.

Effect of Motion. The packing or seal will behave differently according to the direction of movement, whether pumping or motoring. For instance, in a pump gland the ram moves inwards against pressure, so that the inward movement of the ram tends to oppose leakage and extrusion; in a hydraulic press gland, the press ram moves outwards under pressure, and the friction of the ram assists extrusion of the packing and the movement of the ram tends to assist leakage. With "chevron"-type and S.E.A gland packings, a pump ram tends to unwedge the packing on the pumping stroke, whereas a press ram tends to wedge the packing on the working stroke.

The conditions are reversed for a piston packing. Thus, with a piston packing of a hydraulic press, the friction of the packing within the cylinder bore resists extrusion on the working stroke,

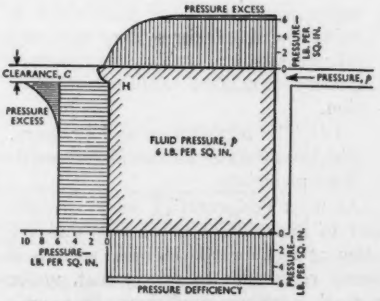

Fig. 15. Stress Distribution across the Faces of a Seal

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and in a piston packing of a pump, the friction between the packing and the cylinder bore assists extrusion on the delivery stroke.

Synthetic Rubber. The term "synthetic rubber" covers a material having widely varying properties such as hardness, resilience, cold flow, aging, etc.; these properties vary with the temperature, pressure, and nature of the liquid being sealed.

Packaging of Synthetic Rubber Seals. McCuiston, Clark, Clark and Cheyney (1948) have reported that the "preaging" of packings in hydraulic fluid gave much more consistent results than packings that had not been "aged," or those which had been subjected to a variable amount of air aging. They suggested that the packing should be soaked in hydraulic oil, wrapped in cellophane, pliofilm or aluminum foil and packaged in a suitable container, the latter to be stamped with the appropriate identification and date of manufacture.

Friction and Wear. The static friction increases with time (see Fig. 16) and the rate of increase will be dependent on the following conditions:—

(a) The physical relationship between the surfaces: clearance, interference, resilience and surface finish.

(b) The chemical relationship between the surfaces: some synthetic rubbers have a tendency to become bonded to certain metals.

(c) The chemical and physical relationship between the fluid which is being sealed and the sealing surfaces; oil and synthetic rubber may form a gummy compound resulting in adhesion.

(d) The temperature and pressure, which will affect all the above conditions materially.

As most reciprocating seals are subject to intermittent movement, and are often at rest for considerable periods it seems reasonable to expect that, under normal working conditions, the static friction will have the greatest influence

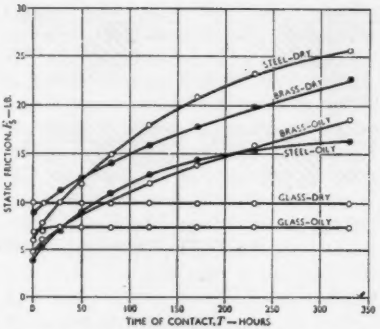

Fig. 16. Increase in Static Friction with Time of Contact for Three Materials

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on the life of a hydraulic seal. Some research workers have been inclined to relate the life of a seal to the kinetic friction and the number of *continuous* reciprocations; their results would be very different if they were to allow one hour to elapse between each stroke, making four reciprocations per day. The author maintains that continuous reciprocation will give no true measure of the life of a hydraulic seal under normal working conditions because, when reversal is instantaneous, the static friction does not come into effect. It is probably the breaking of adhesion at the commencement of each stroke which causes most of the damage to the seal.

The ideal conditions to ensure long life would be a low coefficient of friction combined with a low rate of increase of static friction. Such conditions can be assessed in the laboratory in a matter of hours; whereas, to assess the life of a hydraulic seal by intermittent reciprocation, such as described above, would take months or even years.

Novel types of Hydraulic Seals. It would be impossible to enumerate all the novel types of hydraulic seal which have been invented: one invented by J. H. F. Stewart is shown in Fig. 17. According to Stewart, the ideal seal should have a very narrow contact edge

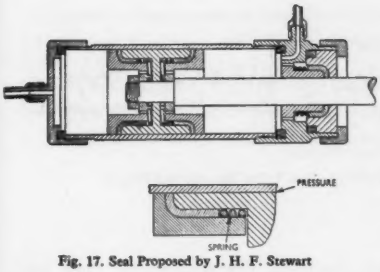

Fig. 17. Seal Proposed by J. H. F. Stewart

adjacent to the pressure, and this is achieved by causing the synthetic rubber to flow round a corner, as shown in Fig. 17*b*, so that the inside edge, which has the shortest distance to travel, is pressed most firmly against the surface of the piston. This type of seal is also being used with considerable success in America as a shaft seal for aircraft pumps; it is probably better known as a shaft seal than as a reciprocating seal. Its advantage is that it is caused to flow inwards, by fluid pressure augmented by a spring, so as to compensate for wear: it will do this even if the wear is greater at one part of the sealed circumference than at another.

Packingless Pistons. The hydraulic seal depends upon a fine lap fit between cylinder and bore. Packingless pistons of comparatively small diameter—up to 2 inches diameter—are suitable for pressures up to 10,000 lb. per sq. in., the diametral clearance between piston and bore being about one micron per inch.* Larger pistons of conventional design are not practicable for higher pressures because the expansion of the cylinder under pressure becomes considerable. Variations of temperature are also a consideration when operating steel in steel with such fine clearances. The rate of leakage is not constant: there is a condition of ebb and flow, and there is usually a reduction of leakage with time.

Packingless pistons are only suitable for use with oil, and the most common application is to packingless piston type control valves, and pumps. Packingless fuel pumps for paraffin and petrol at comparatively low pressures are widely used.

Surface Finish. This is of great importance for packingless pistons subject to high pressure. Fig. 18 is a "Tomlinson" record of the surface finish of the piston and bore of a packingless piston valve, subject to working pressures of about 7000 lb. per sq. in. The radial clearance between piston and bore at no pressure is about one micron, and it increases with the pressure due to the expansion of the valve body. The "Tomlinson" record gives a distorted picture

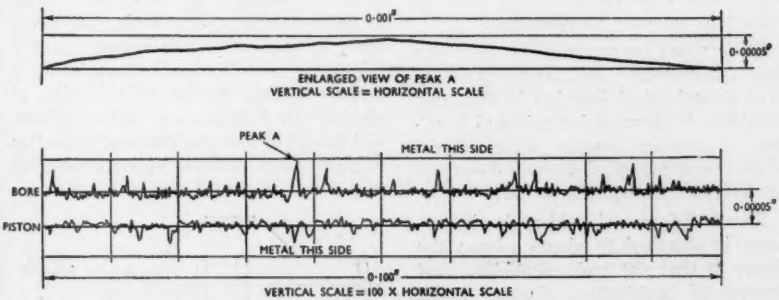

Fig. 18. "Tomlinson" Record of Surface Finish of Piston and Bore of a Packingless Piston Valve

* One micron is equivalent to approximately 40 micro-inches.

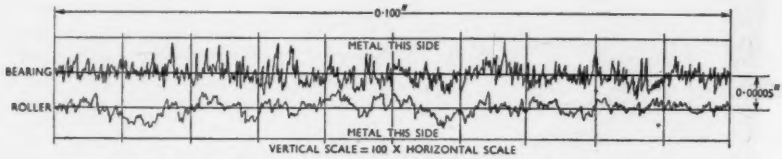

Fig. 19. "Tomlinson" Record of Surface Finish of a High-class Roller Bearing and Roller

of the surface because the vertical scale is one hundred times the horizontal scale. The enlarged view of peak A, where the vertical and horizontal scales are equal, shows that the surface is undulating, and that the maximum depth of undulation is one micron. For comparison a "Tomlinson" record of the surface finish of a high-class roller bearing and roller is shown in Fig. 19.

It is desirable that the surface finish should be specified when the leakage past packingless pistons and the effect of friction are being investigated.

The Grooving of Packingless Pistons. The effect of a series of annular grooves in either the spindle or the bore, or both, and the effect of varying the clearance between spindle and bore, are deserving of considerable research. These effects should be studied both for packingless pump rams on high-speed pumps in which the delivery stroke may be made in one-fiftieth of a second, and packingless piston type valves which may be subject to pressure continuously.

Quite apart from any effect they may have as a hydraulic seal, annular grooves have a mechanical function in that they provide a lodgement for grit and particles of foreign matter which might otherwise cause axial grooves down the full length of the spindle or bore.

Sharp Edges. Considerable importance is attached to sharp edges: the theory is that the ends, shoulders, and grooves of packingless pistons should have sharp edges because round corners tend to entrain small particles of grit and foreign matter between the piston and the bore. The author saw some fuel pump plungers in America which were chromium plated on a stainless steel core, as shown in Fig. 20. Before chroming, the spindle was reduced to contain the chrome so that in the final grinding the sharp edge was produced on the stainless steel; if the chrome had extended to the edge, it might have peeled off and the edge might have become ragged.

It may be found that clean sharp edges are equally important for synthetic rubber seals. Ragged edges may be expected to encourage leakage.

Hydraulic Lock. At pressures exceeding 2000 lb. per sq. in., hydraulic lock becomes an important phenomenon: a valve spindle subject to a certain pressure for a period of time will become locked. Pressure and time appear to be the determining factors: up to a certain pressure the valve does not lock; then at a higher pressure the valve will lock when subjected to pressure for twenty or thirty seconds, and when the pressure is released it will unlock after a minute or two. Hydraulic lock is not a definite seizure, and there is no sign of picking up between the valve spindle and the bore. The phenomenon is similar to that of the increase of static friction

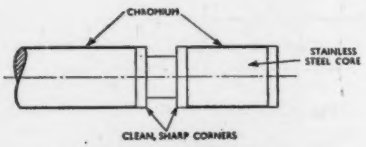

Fig. 20. Fuel-pump Plunger

in hydraulic seals, and is worthy of research; other factors, such as oil viscosity, will affect the phenomenon.

Acknowledgment. Figs. 15 and 16 are reproduced from the Ministry of Supply Scientific and Technical Memorandum No. 3/47 with the permission of the Controller, His Majesty's Stationery Office. Fig. 14 is reproduced from *Product Engineering.*

APPENDIX

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ENGINEERING STEELS UNDER COMBINED CYCLIC AND STATIC STRESSES

ACKNOWLEDGMENT

This article constituted the presidential address delivered on 21 October 1949 to the Institution of Mechanical Engineers by H. J. GOUGH, C.B., M.B.E., D.Sc., F.R.S. It discusses the design of engineering components subject to combined flexual and torsional stresses, the combination of reverse bending stresses, reverse torsional stresses, static bending stresses, and static torsional stresses. It further explains the modifying influences of oil holes, sharp transition fillets, splines, etc., associated with practical design. The address appeared in the Proceedings of the Institution of Mechanical Engineers for 1949.

INTRODUCTION

The customary theme for a Presidential Address is a general survey or a detailed description of one or more fields of engineering in which the speaker has been engaged. In the 1946* Thomas Hawksley Lecture,[1] I was privileged to present an account of one aspect of my war work with the three Services and, in a Centenary Lecture[2] given in the following year, to refer to mechanical engineering in the industry in which I am now engaged. It may, therefore, be not inappropriate to turn to an item of personal engineering research and discuss the behavior of some engineering steels under combined fatigue stresses.

Those concerned with the practical design of engineering components subjected to combined flexural and torsional stresses, of which the engine crankshaft is the typical example, are aware of the extreme scarcity of available experimental data which are of direct application. In the stressing system to which such components are subjected, at least four variables, singly and in combination, are of importance: reversed bending stresses, reversed torsional stresses, static bending stresses, and static torsional stresses. Valuable and essential as such information would be, a knowledge of the resistance of smooth cylindrical shafts to these combined stresses is not sufficient; in practice, the modifying influence of such features as grooves, oil-holes, sharp transition fillets and splines, etc., unavoidably associated with practical design, must also be understood as these so often determine the life of the component.

A long-range investigation into this subject was, therefore, planned for which new types of high-speed testing machines had to be developed. It was decided to divide the investigation into two stages, the first of which the two cyclic vari-

* A list of references in numerical order is given in the Appendix.

ables only would be studied; in this stage, a general survey has been made of the behaviors of a representative selection of engineering steels, covering a wide range of compositions, heat-treatments, and tensile strengths. With the exception of some tests on a mild steel by Stanton and Batson,[3] no previous work had been carried out on combinations of these fatigue stresses. Apart from time spent on preliminary development, this first stage of the investigation, which was financed by the Department of Scientific and Industrial Research and in which Mr. H. V. Pollard, A.M.I.Mech.E., was my collaborator, was started towards the end of 1933; interim reports, giving results obtained with two steels and one cast iron, were published by the Insitution in 1935[4] and 1936[5]. The second stage of the investigation, in which all four variables were studied in any desired combination, was commenced in 1936 with the assistance of Mr. W. J. Clenshaw, B.Sc., A.M.I.Mech.E., as joint investigator; one material, a 65-ton aircraft alloy steel, was selected for an extensive series of tests which was financed by the Air Ministry. Both parts of the investigation were actively pursued until the outbreak of war brought the investigation to an end; the experimental programme was completed but detailed analysis of the results and their publication had to be deferred to the post-war period. The complete account of the investigation, with supplementary mechanical tests, metallurgical examination of materials, etc., is too extensive for an Institution paper; it is in course of publication as a monograph[6]. The reference I shall make tonight must be confined to an indication of the field covered and the nature of some of the principal conclusions reached; those specially interested in the subject will, no doubt, consult the full account. I hope, also, that what I have to say may draw the attention of other investigators to the wide field of engineering materials and stress combinations that remains to be explored.

Now, in this experimental investigation into combined fatigue stresses, small specimens, not exceeding half an inch in diameter, have been employed, tested at cyclic speeds of 1500–2200 per minute. I hope to show that the results yield simple relations which should be of practical use in affording a basis for design; these relations, for any required material, only require simple fatigue data such as are available in the literature or can be easily determined experimentally. To what extent can they be applied to full-sized components, operating at other speeds?

The influence of *speed effect* appears to be fairly well established. The results of many investigations have shown that, in general, the fatigue range increases with cyclic frequency to an extent dependent on the material under test; with many metals, the increase is appreciable when frequencies of the order of 10,000 cycles per minute are reached and, at very high frequencies, marked further increases in the fatigue range have been recorded. But it can be concluded that, at frequencies between 150 and 5,000 cycles per minute, speed effect may be neglected. Much more work, however, requires to be done on *size effect* in fatigue. In studying this very important problem, it is most desirable, but very difficult, to ensure that specimens of the same material, tested in various sizes, have, initially, identical physical properties, while the use of geometrically similar specimen forms and loading conditions is also essential. A number of investigations, designed to meet these conditions, have been made by Peterson,[7, 8] Mailander and Bauersfeld,[9] Horger,[10] Dolan and Hanley,[11] and others. These show quite definitely that, when discontinuities of section are present, the fatigue range decreases, but at a decreasing rate, as the specimen size increases; the results also suggest

that the fatigue strengths probably attain minimum values when a specimen diameter of about 2 inches is reached, but, as the sizes which have been systematically investigated rarely exceeded 3 inches, further tests on larger sizes are required. In some investigations, notably those of Peterson and Wahl,[12] the practical reductions in strength, resulting from the presence of some form of discontinuity of section in a series of geometrically similar specimens, have been compared with the theoretical effect of the same discontinuity as determined by the methods of mathematical analysis, photoelasticity, static strain-measurement, etc. In general, the full theoretical weakening effect is not realized, the extent of divergence between theory and experiment varying with the material under test. In some cases, the approach is fairly near, particularly with some of the high-tensile steels; with other steels, the actual reduction of strength falls considerably short of that predicted by theory. Safe design could, no doubt, be based on theoretical stress concentration factors but, in many cases, it would not be economical design. In regard to smooth specimens free from discontinuities of section, the available evidence is contradictory: some investigators have reported no size effect, up to a specimen size of about 3 inches; others find the same effect as with specimens containing discontinuities. Of course, in all such comparative tests on a practical engineering metal or alloy, there is one factor—grain size—which is usually kept constant in order to satisfy the requirement of identical physical properties, but, except for this departure from similarity in test conditions, it is difficult to understand why a scale effect should exist under geometrically similar conditions of form and loading. Clearly, the subject of size effect in fatigue is of real importance to engineering design; it was hoped to supplement the present combined fatigue program by a few carefully selected tests using much larger specimens of, say, 2½ inches in diameter, but this must be left to future investigators.

STAGE 1. COMBINED FATIGUE STRESSES, EMPLOYING TWO INDEPENDENT CYCLIC VARIABLES

The Testing Machine. The type of fatigue machine—the No. 1 Combined Stress Fatigue Testing Machine—developed for this stage of the investigation has been fully described[4] in the 1935 paper to the Institution; it will be sufficient to recall its method of operation. A battery of four such machines was constructed and used continuously throughout the research. As indicated in Fig. 2, one end of the specimen is attached riddidly to frame of the machine. A synchronous motor drives a vertical disk carrying out-of-balance weights, of adjustable mass, which produce an alternating vertical load, $\pm P$. This alternating load is applied to the specimen through a horizontal loading arm, which

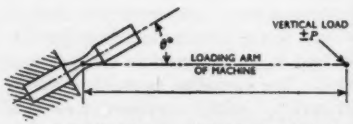


Fig. 2. Loading System applied to Specimen in the No. 1 Combined Stress Machine

makes an angle θ with the axis of the specimen; θ can be set at any required angle from 0 to 90 deg. The alternating bending and alternating torsional moments thus applied to the specimen are proportional to cos θ and sin θ, respectively. The axle of the rotating disk is carried by a horizontal system of four cantilever springs; the speed of the disk is carefully adjusted to be equal to the resonant frequency of all the moving

parts vibrating on the outer ends of these springs. All inertia forces, other than the vertical forces due to the out-of-balance weights, are thereby eliminated from the specimen. Thus, with the standard cylindrical specimen, when the setting angle of the machine is $\theta = 0$ deg., cycles of reversed plane bending stress, $\pm f$, are applied; and cycles of reversed torsional stress, $\pm q$, when $\theta = 90$ deg.; for any intermediate setting, $f/q = 2 \cot \theta$. The fatigue range of each material has been determined, by endurance tests on a 10^7 cycles basis, at each of seven settings, i.e. $\theta = 0$, 15, 30, 45, 60, 75, and 90 deg. The cyclic stresses are in phase: the operating frequency of the machine is 2130 cycles per min.

Test Materials and Program. Crankshaft steels used in practice cover a wide range of tensile strengths, obtainable by a considerable variety of chemical compositions and heat-treatments; some difficulty was encountered in selecting a representative, but necessarily limited, series for test purposes which would have practical interest and also afford some information regarding the effect, if any, of some of these varations. It was decided: (1) to choose a series of standard steels each representing a definite class in regard to tensile strength, in some cases including a pair of steels in the same strength class in order to examine the effect of chemical composition; (2) to include four types of structure, namely: (*a*) ferrite and pearlite, as in normalized carbon steels, (*b*) pearlite, (*c*) uniform matrix containing spheroidized cementite, as in fully annealed steels of moderate carbon content, and (*d*) hardened and tempered; (3) to test one steel in two forms of heat-treatment having widely differing impact strengths but the same tensile strength. This gave steels of ten chemical compositions or, as two steels were tested in

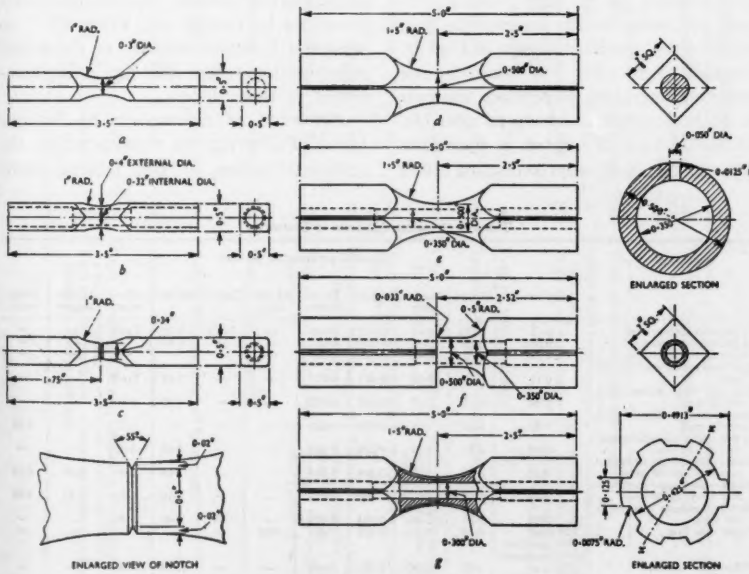


Fig. 3. Combined-fatigue-stress Specimens

each of two forms of heat-treatment, twelve test materials, each of which is an essentially ductile steel. By contrast, two cast irons were also included "Silal" chosen to represent a near approach to a brittle material, and "Nicrosilal" to represent a comparatively ductile cast iron.

The programme of combined stress fatigue tests which has been made on these fourteen materials is as follows. Using the type of solid specimen shown in Fig. 3*a*, the fatigue strength of each material has been determined at each of the selected seven values of θ (0, 15, 30, 45, 60, 75, and 90 deg.). To investigate the effect of specimen form, a similar complete series of fatigue determinations has also been made on each of two steels, using hollow specimens as in Fig. 3*b*. Tests to determine the effect of one form of discontinuity of section on the resistance to combined fatigue stresses have been made on seven of the steels. The discontinuity was produced by machining a 55 deg. "sharp" V-notch circumferentially round the test section of the specimen shown in Fig. 3*c*. Although every care was taken in the machining operation to produce the bottom of each notch as sharp as possible, each notch had, of course, a transition curve at its root, approximating to a "root radius." Using a projection method, this "root radius" of each specimen was measured and recorded; it was remarkably constant for any individual steel but varied somewhat from steel to steel: with most steels the notch radius was round about half a thousandth of an inch; with others, it was round about one-thousandth.

The essential particulars of the test materials and the combined fatigue stress program only are summarized in Table 1, in which the actual tensile and notched bar values are inserted: the chemical analyses are as stated in Table 2. The results of some of the usual mechanical test data relating to the test materials are summarized in Table 3. In regard to items 1, 8, and 13 of Table 1, the results of the tests made on solid and hollow specimens of the 0.1 percent carbon steel (normalized) and the 3½ percent nickel-chromium steel (normal impact), together with the tests made on solid specimens of "Silal" cast iron, formed the subject of interim reports previously presented by Gough and Pollard,[4, 5] to present a balanced account of the whole investigation, they will again be mentioned in this address.

Accuracy of Estimation of Fatigue Limit. Following the usual practice, the recorded values of the fatigue limits

TABLE 2. CHEMICAL ANALYSES OF MATERIALS

Material	Chemical composition, per cent										
	Carbon	Silicon	Manganese	Sulphur	Phosphorus	Aluminium	Copper	Nickel	Chromium	Molybdenum	Vanadium
0·1 per cent carbon steel	0·12	0·185	0·61	0·012	0·016	—	0·075	0·06	Trace	—	—
0·4 per cent carbon steel	0·39	0·23	0·65	0·017	0·02	—	—	0·12	—	—	—
0·9 per cent carbon steel	0·86	0·05	0·13	0·026	0·02	—	—	—	—	—	—
3 per cent nickel steel (30–35 tons)	0·11	0·22	0·42	0·011	0·011	—	—	3·18	Trace	—	—
3–3½ per cent nickel steel (45–50 tons)	0·34	0·17	0·57	0·007	0·011	—	—	3·25	0·06	—	—
Chromium-vanadium steel (45–50 tons)	0·41	0·23	0·71	0·006	0·017	—	—	—	1·27	—	0·28
3½ per cent nickel-chromium steel (55–60 tons)	0·30	0·21	0·58	0·006	0·018	—	—	3·62	0·85	—	—
Nickel-chromium-molybdenum steel (60–70 tons)	0·24	0·20	0·57	0·004	0·015	—	—	3·06	1·29	0·54	0·25
Nickel-chromium-molybdenum steel (75–80 tons)	0·24	0·27	0·57	0·007	0·011	—	—	3·10	1·33	0·41	0·25
Nickel-chromium steel (95–105 tons)	0·28	0·23	0·48	0·004	0·018	—	—	4·42	1·36	—	—
"Silal" cast iron	2·09 total (1·99 graphitic) (0·10 combined)	6·39	1·13	0·031	0·047	0·08	—	—	—	—	—
"Nicrosilal" cast iron	1·88 total (1·51 graphitic) (0·37 combined)	4·25	0·60	0·039	0·033	—	—	17·74	2·39	—	—

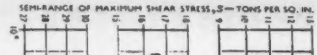
TABLE 1. MATERIALS AND PROGRAMME

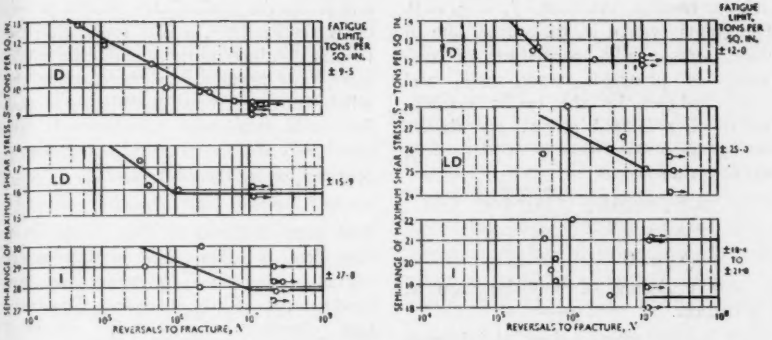
Item	Description of material	Heat-treatment	Form of microstructure	Tensile strength, tons per sq. in.	Izod value, ft.-lb.	Special objects and C.S. fatigue tests
1	0·1 per cent carbon steel (normalized)	Normalized from 900 deg. C.	Ferrite and pearlite	27·9	89	Tests on solid and hollow specimens
2	0·4 per cent carbon steel (normalized)	Normalized from 850 deg. C.	Ferrite and pearlite	42·0	30	Tests on solid and notched specimens } A steel tested in each of two heat-treated conditions, to reveal effect of different forms of structure of the same chemical composition
3	0·4 per cent carbon steel (spheroidized)	Normalized from 900 deg. C.: heated for six days at 650 deg. C. and slowly cooled	Uniform matrix containing spheroidized cementite, as in fully annealed steels of moderate carbon content	30·9	32	Tests on solid specimens }
4	0·9 per cent carbon steel (pearlitic)	Normalized from 820 deg. C.	Pearlite	54·9	3½	Tests on solid specimens
5	3 per cent nickel steel (30–35 tons)	Oil-hardened from 850 deg. C.: tempered 700 deg. C.: air cooled	Hardened and tempered	34·1	100	Tests on solid and notched specimens
6	3–3½ per cent nickel steel (45–50 tons)	Oil-hardened from 850 deg. C.: tempered 610 deg. C.: air cooled	Hardened and tempered	46·8	86	Tests on solid and notched specimens } Two steels selected as having the same tensile strength but differing chemical composition
7	Chromium-vanadium steel (45–50 tons)	Oil-hardened from 850 deg. C.: tempered 700 deg. C.: air cooled	Hardened and tempered	48·7	96	Tests on solid and notched specimens }
8	3½ per cent nickel-chromium steel (normal impact)	Oil-hardened from 830 deg. C.: tempered 620 deg. C. and cooled in water	Hardened and tempered (quickly cooled)	58·0	76	Tests on solid, hollow, and notched specimens } The same steel in the normal and embrittled conditions: same tensile strength but widely differing notched bar values
9	3½ per cent nickel-chromium steel (low impact)	Oil-hardened from 830 deg. C.: tempered 1½ hours at 620 deg. C. and cooled to 100 deg. C. in 42 hours	Hardened and tempered (slowly cooled)	58·1	5½	Tests on solid and notched specimens }
10	Nickel-chromium-molybdenum steel (60–70 tons)	Normalized 900 deg. C.: oil-hardened from 850 deg. C.: tempered 640 deg. C.: air cooled	Hardened and tempered	64·8	90	Tests on solid specimens
11	Nickel-chromium-molybdenum steel (75–80 tons)	Oil-hardened from 850 deg. C.: tempered 600 deg. C.: air cooled	Hardened and tempered	80·5	55	Tests on solid and notched specimens
12	Nickel-chromium steel (95–105 tons)	Air-hardened from 820 deg. C.: tempered 200 deg. C.: air cooled	Hardened and tempered	108	24	Tests on solid specimens
13	"Silal" cast iron	As cast	Cast	14·9	½	Tests on solid specimens. Chosen as a near approach to a brittle material
14	"Nicrosilal" cast iron	As cast	Cast	14·2	1½	Tests on solid specimens. Chosen as representing a comparatively ductile cast iron

TABLE 3. SOME MECHANICAL PROPERTIES OF THE TEST MATERIALS

Mechanical properties	Materials													
	0·1 per cent carbon steel (normalized)	0·4 per cent carbon steel (normalized)	0·4 per cent carbon steel (spheroidized)	0·9 per cent carbon steel (pearlitic)	3 per cent nickel steel (30–35 tons)	3–3½ per cent nickel steel (45–50 tons)	Chromium vanadium steel (45–50 tons)	3½ per cent nickel-chromium steel (normal impact)	3½ per cent nickel-chromium steel (low impact)	Nickel-chromium-molybdenum steel (60–70 tons)	Nickel-chromium-molybdenum steel (75–80 tons)	Nickel-chromium steel (95–105 tons)	"Silal" cast iron	"Nicrosilal" cast iron
Tensile limit of proportionality, tons per sq. in.	12·5	14·5	14·9	7·7	15·8	26·0	31·5	38·3	13·5	47·0	21	24	1·9	1·9
Upper yield point in tension*, tons per sq. in.	16·6	23·4	18·6	N.D.	N.D.	38·0	44·1	49·2	48·9	61·3	N.D.	N.D.	N.D.	N.D.
Ultimate tensile strength, tons per sq. in.	27·9	42·0	30·9	54·9	34·1	46·8	48·7	58·0	58·1	64·8	80·5	108	14·9	14·2
Elongation at fracture ($l = 4\sqrt{A}$), per cent	40½	31	39½	13½	40½	26½	26½	25	24	23½	20½	16	½	1
Reduction of area at fracture, per cent	70	58½	67	18	72½	67½	66	65	60½	67	60	52	0	1·3
Young's modulus, E, lb. per sq. in. $\times 10^{-6}$	28·8	30·0	30·1	29·3	29·2	29·2	30·3	29·0	28·9	29·1	29·2	30·0	17·5	19·6
Brinell hardness number (2 mm. ball, 120 kg. load)	127	195	144	248	165	237	229	282	278	325	394	479	322	182
Torsional limit of proportionality, tons per sq. in.	8·5	10·5	5·2	7·5	9·0	19·0	24·5	23·7	18·3	31·4	26·6	28·8	4·4	2·3
Yield point in torsion*, tons per sq. in.	11·7	16·8	11·7	N.D.	N.D.	27·8	30·8	37·8	N.D.	46·3	59·1	N.D.	N.D.	N.D.
Torsional modulus of rupture, tons per sq. in.	32·7	41·7	36·3	48·9	38·0	44·3	46·2	52·3	51·2	57·3	67·5	89·2	23·8	32·5
Total twist at fracture ($L/d = 8\frac{1}{2}$), deg.	2,859	1,600	2,190	552	2,014	1,991	1,402	1,230	1,150	2,085	1,066	630	8	170
Modulus of rigidity, G, lb. per sq. in. $\times 10^{-6}$	11·5	11·8	11·8	11·4	11·5	11·4	12·1	11·4	11·5	11·5	11·6	11·3	7·6	8·6
Izod notched bar value, ft.-lb.	89	30	32	3½	100	86	96	76	5½	90	55	24	½	1½

* N.D. denotes "no definite yield point".





O→ DENOTES SPECIMEN UNBROKEN

D Fatigue limit definitely indicated (225 results).
LD Fatigue limit less definitely indicated (34 results).
I Fatigue limit indefinitely indicated (7 results).

Fig. 4. Typical *S/N* Curves

have been determined by inspection of the S/N curves. It is unnecessary and impracticable to present in the present address, either in graph or tabulated form, the detailed results of the 266 separate fatigue determinations which have been made, but it is felt that some indication may usefully be given of the accuracy of the values quoted. The accuracy of the estimation is determined entirely by the degree of "scatter" of the results, which is a reflection of the uniformity of the test material, from specimen to specimen. Some materials exhibited a very high degree of uniformity; a few, usually the high-tensile, heat-treated alloy steels, or cast materials, sometimes less satisfactory *S/N* curves. The accuracy of the testing machine was extremely high. The following method has been adopted. The *S/N* curves have been divided into three classes, denoted by "D" (fatigue limit *definitely* indicated), by "LD" (*less definitely* indicated), or by "I" (*indefinitely* indicated). Fig. 4 shows three pairs of actual *S/N* curves to illustrate these three classes: each pair represents the limits of the class. An analysis showed that of the total of 266 *S/N* curves, 225, or 85 percent, were "D", 34, or 12½ percent, were "LD", while 7 only, or 2½ percent, were "I". In the graphs to be presented, the appropriate classification of each individual test point is indicated.

Stress Nomenclature. We are concerned only with the fatigue limit (i.e., the endurance limit at 10^7 cycles) at any applied combination of flexural and twisting moments; it will be sufficient, and most convenient for application to design, to express each fatigue limit in terms of the maximum bending and torsional stresses induced in the specimen. Let

$\pm f$ = range of plane bending stress due to the cyclic flexural moment;

f_s = constant plane bending stress due to the superimposed static bending moment;

$\pm q$ = range of shearing stress due to the cyclic twisting moment; and

q_s = constant shearing stress due to the superimposed static twisting moment.

Then the imposed bending stress system is $(f_s \pm f)$ and the imposed shearing stress system is $(q_s \pm q)$. In all tests made in the No. 1 combined stress

fatigue testing machine, $f_s = q_s = 0$. The stress system is, of course, one of plane stress only, consisting of two ranges of principal stress, $\pm p_1$ and $\pm p_2$, induced by the cyclic loading, the third principal stress, p_3, always havings zero value as it occurs at a free surface, thus:—

$$\pm p_1 = \tfrac{1}{2}f + \sqrt{\tfrac{1}{4}f^2 + q^2}$$
$$\pm p_2 = \tfrac{1}{2}f - \sqrt{\tfrac{1}{4}f^2 + q^2}$$
$$p_3 = 0$$

p_1 and p_2 being of opposite sign at any instant.

It is convenient to assign special terms to the cases of simple reversed stresses of each kind when acting alone as these are "end-points" to which constant reference will be made. Let

$\pm b$ = fatigue limit under reversed plane bending stresses only, and

$\pm t$ = fatigue limit under reversed shearing stresses only.

The f/q Relation. With a very few exceptions,[3, 13], no tests have previously been reported on the fatigue resistance of metals over the complete range from reversed bending stresses to reversed torsional stresses with intermediate combinations (the present investigation was designed to assist in filling the gap), but considerable data exist concerning the values of $\pm b$ and $\pm t$ for a wide range of metals and alloys, and it is established that the ratio of t/b varies considerably; an examination [14] of forty-eight such results showed ratio values varying from 0.44 to 0.80. Hence, before the present research was commenced, it was already evident that no general f/q relation for combined fatigue stresses could emerge consistent with any criterion for elastic failure which had been advanced*, if the criterion postulated a single-valued relationship between b and t. In the preliminary results previously reported [4, 5], it was found that the results obtained with two ductile steels gave very regular f/q curves over the whole range of stress combinations and, for each material, experimental data fitted very closely to an *ellipse quadrant* represented by the expression:—

$$f^2/b + q^2/t^2 = 1 \quad . \quad . \quad (1)$$

The data obtained in tests made on a cast iron (Silal) also plotted fairly regularly, but the f/q curve was certainly not an ellipse quadrant. The fracture characteristics of this material strongly suggested that principal stress or principal strain conditions were of importance but the f/q curve, although similar in shape, lay intermediate between curves given by the two-dimensional representation of those two criteria and so could not be definitely associated with either. In 1937,[15] with Mr. Pollard, I reported a separate investigation of four cast-iron crankshaft materials. The f/q curves obtained were of the same general geometrical shape to that previously given by Silal; also, the fracture planes of these four irons were in good agreement with those subjected to maximum principal stress (or strain). Apparently, these five cast materials conformed consistently to some general form of f/q relation (although the values of b/t varied quite widely) which was different from the ellipse quadrant found for the ductile steels. It was shown that the several f/q relations for the cast irons were all represented closely by the general form of *ellipse* arc†, expressed by:—

$$\frac{q^2}{t^2} + \frac{f^2}{b^2}\left(\frac{b}{t} - 1\right) + \frac{f}{b}\left(2 - \frac{b}{t}\right) = 1 \quad . \quad (2)$$

The ellipse quadrant and ellipse arc relations were thus derived. They were regarded at the time as empirical, having

* This aspect has been fully discussed previously[4]. There is, of course, no *a priori* reason why elastic failure and fatigue fracture should be subject to any common stress criterion, as the two phenomena differ so essentially, especially when cyclic stresses other than reversed stresses are in operation.

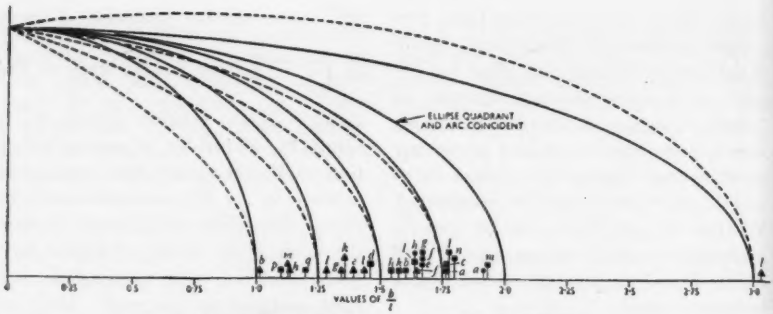
† The derivation of this general case of a conic to cover certain limiting values of b/t has been fully described.[15]

been chosen merely because these two simple geometrical forms fitted fairly closely those experimental data for the two, apparently distinct, groups of "ductile" steels and "brittle" cast irons which were then available; given any experimental "end-point" values of b and t, the curves can be constructed. Whether or not they were of general application could not be considered until the results of the complete investigation became available.

However, in discussing the first interim report,[4] Stanfield suggested that a clue to a physical explanation of the form of the f/q relation might be obtained by assuming, with Perry, that the resistance to shear involves an "internal frictional resistance" whose value is directly proportional to the normal stress acting across the shear plane. The normal resistance to shear is assumed to be increased by the frictional resistance, failure occurring when the sum of the two terms reaches a limiting value given by $(S + \mu p)$ or $(\lambda S + \mu p)$; the planes of failure are not principal planes, but as determined by a relation of the type, $\alpha p_1 - \beta p_2 =$ constant, where p_1 and p_2 are the principal stresses. On this assumption, and by choosing appropriate values of the constants, Stanfield obtained an expression which is mathematically identical with the above equation (2), for the ellipse arc. Stanfield concluded that his expression fitted the "Silal" data satisfactorily but did not fit the data for the ductile steels so closely as the ellipse quadrant. During the same discussion [4] and, later, in a separate paper [18], Professor Guest showed that his criterion for failure of elasticity under static stressing, i.e. $(p_1 - p_2) + \lambda(p_1 + p_2) = C$, can be rewritten to give exactly the expression given in equation (2) above, by taking the value of his constant λ equal to $\left(\frac{2t}{b} - 1\right)$. The second term of Professor Guest's relation represents, as originally defined by its author, a fraction of the volumetric stress and is not, necessarily, to be associated with the Perry conception of internal friction; the value of λ, while a constant for a given material, was assumed to vary from material to material. More recently, Mr. H. L. Cox, of the National Physical Laboratory, has made a mathematical investigation [17] into the effect of flaws and elliptical holes on the strength of materials subjected to complex stress systems. He obtains a relation between the bending and shearing stress components which, again, is represented by equation (2).

Thus, the present position in regard to the ellipse quadrant and the ellipse arc, adopted initially as convenient geometrical forms which fitted the early results, is that, while no physical interpretation has yet been advanced for the ellipse quadrant, three separate explanations exist, based on quite different physical bases, any one of which would be consistent with the representation of the form of the f/q relation by the ellipse arc. It will, of course, require some future experimental investigation of quite a different nature—but in which the determination of the influence of the third principal stress, p_3, must certainly be included—to determine whether the fatigue of metals under combined stresses is, in fact, governed by any of the physical bases on which these three explanations rest.

As each series of experimental data obtained will be compared with the ellipse quadrant and arc, it may be helpful first to show how the divergence of



——— Ellipse quadrant. — — — — Ellipse arc.

■ Solid specimen
● Hollow specimen
▲ Specimen with discontinuity of section

} $f_s = q_s = 0$ throughout.

a 0·1 per cent carbon steel (normalized).
b 0·4 per cent carbon steel (normalized).
c 0·4 per cent carbon steel (spheroidized).
d 0·9 per cent carbon steel (pearlitic).
e 3 per cent nickel steel (30–35 tons).
f 3–3½ per cent-nickel steel (45–50 tons).
g Chromium-vanadium steel (45–50 tons).
h 3½ per cent nickel-chromium steel (normal impact).
k 3½ per cent nickel-chromium steel (low impact).
l Nickel-chromium-molybdenum steel (60–70 tons).
m Nickel-chromium-molybdenum steel (75–80 tons).
n Nickel-chromium steel (95–105 tons).
p "Silal" cast iron.
q "Nicrosilal" cast iron.

Fig. 5. The Dependence on the Ratio Value b/t of the Divergence between an Ellipse Quadrant and the Corresponding Ellipse Arc

any corresponding pair of curves drawn through the same "end points" depends essentially on the value of the ratio b/t. In Fig. 5, six corresponding pairs of curves have been drawn for b/t ratio values of 1, 1.25, 1.5, 1.75, 2.0, and 3.0. At $b/t = 1$ (which is consistent with failure being determined by a constant value of range of maximum principal stress), the quadrant and arc are widely divergent. The divergence decreases progressively as b/t increases until $b/t = 2$ (which is consistent with failure being determined by a constant value of range of maximum shear stress) when the curves are coincident. At still higher values of b/t, the curves cross over relatively, the arc being external to the quadrant, the divergence continuously increasing. All curves meet the b/t axis orthogonally; each quadrant, of course, meets both axes orthogonally; in general, each arc meets the t axis obliquely. The experimental results marked on the diagram refer to tests in which cycles of reversed stresses only were in operation. With the exception of one very special form of specimen where b/t may have, for reasons which will be explained later, the extreme value of 3.04, all experimental values of b/t are spread variously between the limits of 1.0 and 2.0. Thus, in some cases when this divergence is small and non-uniformity of the test material yields irregular test data, definite identification with either the quadrant or the arc is open to some doubt. A critical assessment of the results in this respect has, however, been made, but the data are fully presented to enable these assessments to be independently reviewed.

RESULTS AND CONCLUSIONS

The f/q Relation. The experimental values of the limiting ranges of stress, or fatigue limits, are plotted in Figs. 6 and 7, in the form of f/q diagrams. For each material, the appropriate ellipse quadrant and arc have been drawn through the endpoints,* b and t. Making due allowance for irregularities, each set of data plots in a form that is satisfactorily expressed by one of the two criteria.

(*a*) *Solid Specimen.* The form of specimen used is illustrated in Fig. 3*a*, having a test section diameter of 0.3 inch.

(i) *Ductile Steel (Fig. 6a, b, c, and d).* There are five steels in which the quadrant fits the data very closely and certainly closer than the arc: 0.1 percent carbon (normalized); 3 percent nickel; 3½ percent nickel-chromium (normal impact); 0.4 percent carbon

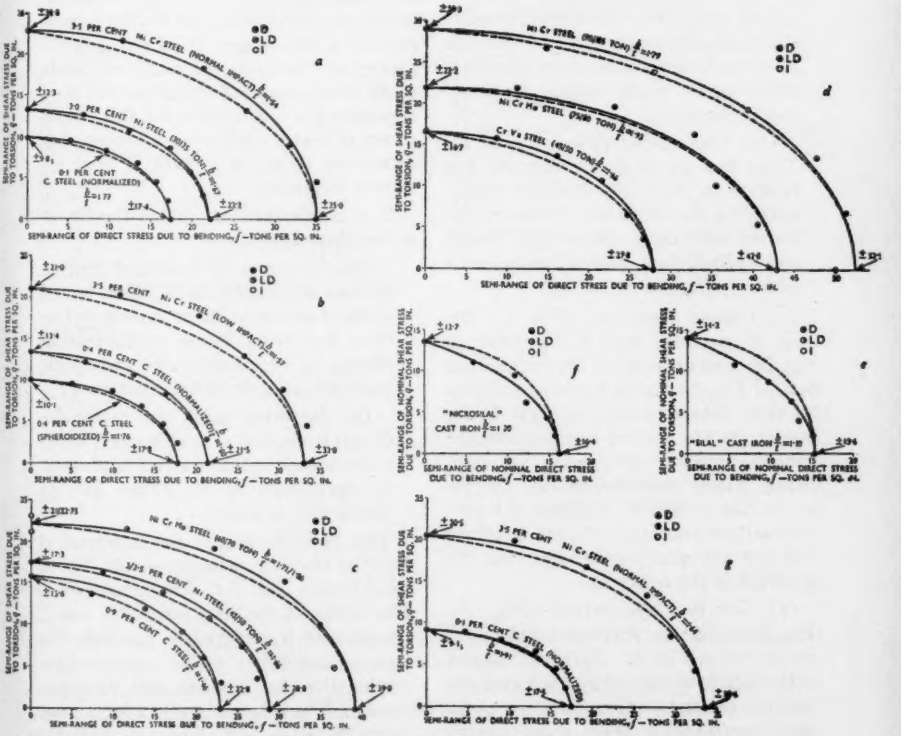

a
b
c
d
e
f
g
●D
●LD
○I
SEMI-RANGE OF DIRECT STRESS DUE TO BENDING, f — TONS PER SQ. IN.
SEMI-RANGE OF SHEAR STRESS DUE TO TORSION, q — TONS PER SQ. IN.
SEMI-RANGE OF NOMINAL DIRECT STRESS DUE TO BENDING, f — TONS PER SQ. IN.
SEMI-RANGE OF NOMINAL SHEAR STRESS DUE TO TORSION, q — TONS PER SQ. IN.
3½ PER CENT Ni Cr STEEL (NORMAL IMPACT)
3 PER CENT Ni STEEL (85/95 TONS)
0·1 PER CENT C STEEL (NORMALIZED)
3½ PER CENT Ni Cr STEEL (FLOW IMPACT)
0·4 PER CENT C STEEL (NORMALIZED)
0·4 PER CENT C STEEL (SPHEROIDIZED)
Ni Cr Mo STEEL (80/95 TONS)
3½ PER CENT Ni STEEL (45/55 TONS)
0·9 PER CENT C STEEL
Ni Cr STEEL (85/95 TONS)
Ni Cr Mo STEEL (90/100 TONS)
Cr V₂ STEEL (45/55 TONS)
"NICROSILAL" CAST IRON
"SILAL" CAST IRON
3½ PER CENT Ni Cr STEEL (NORMAL IMPACT)
0·1 PER CENT C STEEL (NORMALIZED)

Fig. 6. *f/q* Curves

***a*, *b*, *c*, *d*, *e*, and *f* relate to solid specimens; *g* relates to hollow specimens.**

* With one exception only on Fig. 6*d*, where an "LD" value of "b" has been adjusted to fit the other data.

(spheroidized); chromium-vanadium (45-50 tons). There are two steels—the 3½ percent nickel-chromium (low impact) and the nickel-chromium-molybdenum (60-70 tons) — whose data tend to fall slightly outside both curves; the ellipse quadrant is the closer fit. Then there is a group of four steels in which the quadrant and arc are in close proximity and either curve would provide a good fit, but the quadrant is slightly to be preferred: 0.4 percent carbon (normalized); 3-3½ percent nickel (45-50 tons); 0.9 percent carbon; nickel-chromium (95-105 tons). Lastly, for the nickel-chromium-molybdenum steel (75-80 tons), the curves are practically coincident and the data somewhat irregular: even the quadrant affords a conservative design basis, however, and is preferred.

(ii) *Cast Irons (Fig. 6e and f).* There can be no doubt that the f/q relation for each iron is closely represented by the ellipse arc. These results accord with those obtained by Gough and Pollard [15] on four crankshaft cast irons.

(b) Hollow Specimens (Fig. 6). The form of specimen used is illustrated in Fig. 3*b*. Examination of the fatigue test data of Fig. 6*g* shows that the quadrant fits these data very satisfactorily. In the case of the 3½ percent nickel-chromium (normal impact), the curves are sufficiently widely separated to rule out the arc as the criterion; with the 0.1 percent carbon (normalized), the quadrant and arc are nearly coincident, but the quadrant is the better fit.

(c) Grooved Specimens† (Fig. 7). The form of the specimen and notch are shown in Fig. 3*c*. The root diameter at the bottom of the notch is 0.3 inch, the same as the test section diameter of the solid specimens: a direct evaluation can thus be made of the reduction in fatigue strength due to the presence of the notch. If allowance is made for irregular test points, these data, when plotted, give definite f/q curves.

For the 0.4 percent carbon (normalized) and the nickel-chromium-molybdenum (75-80 tons) steels, the arc provides a very close fit to the data, while the arc definitely provides a closer fit than the quadrant for the 3 percent nickel (30-35 tons) and the 3½ percent nickel - chromium (normal impact) steels. The test data of the remaining three steels—3-3½ percent nickel (45-50 tons), the chromium-vanadium (45-50 tons) and the 3½ percent nickel-chromium (low impact)—are irregular and either the quadrant or the arc would provide a satisfactory fit. But a general survey of this group of diagrams leads to the conclusion that the ellipse arc does represent the f/q relation for this whole group of seven steels when a stress concentration set up by a sharp 55 deg. V-groove is present.

It is concluded that, within the limits of the investigation:—

> The resistance to combined fatigue stresses of ductile steels, when tested in the form of solid or hollow cylinders, free from stress concentration effects, is represented by an ellipse quadrant as defined by equation (1).
>
> On the other hand, the resistance of cast irons, and of ductile steels when a stress concentration effect is present, is represented by an ellipse arc as defined by equation (2).

This identification of the geometrical form of the f/q relation should be of considerable use for design purposes. The values of the "end-points," b and t, are all that is required to calculate the fatigue strengths of the intermediate combinations of bending and torsional stresses. The values of b and t for a wide range of metals and alloys are available in the published literature and, if not

† The stresses recorded are, except where otherwise stated, nominal values calculated in the usual manner on the root diameter at the base of the notch and without taking into consideration the stress concentration effect present.

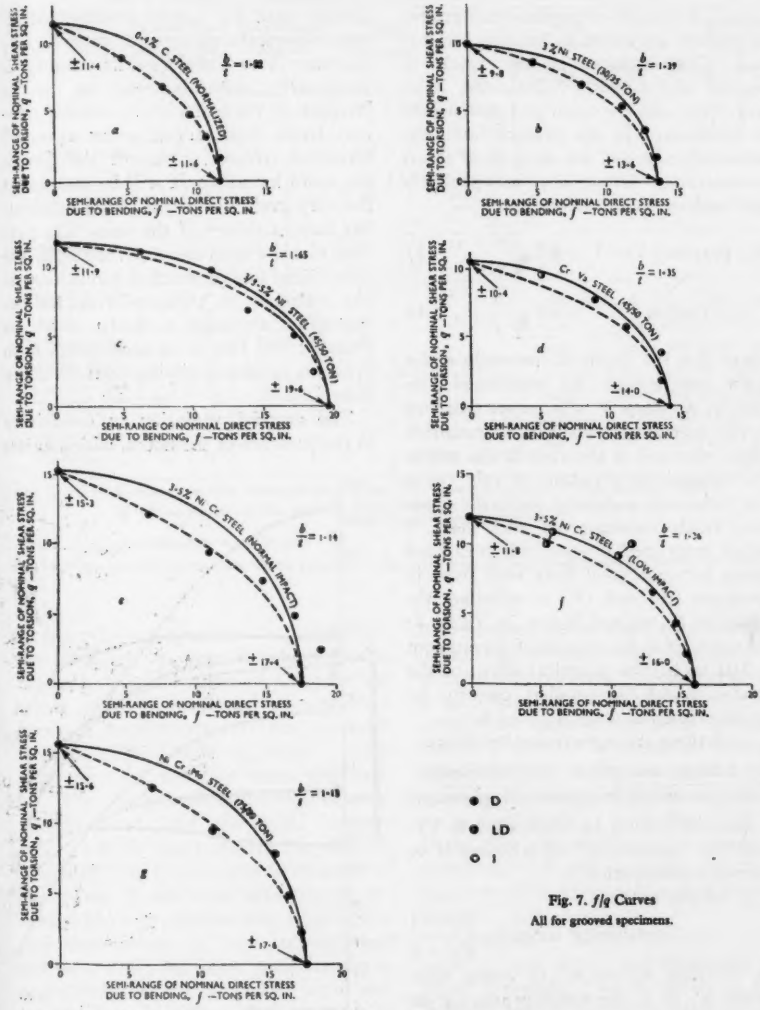

0·4% C STEEL (NORMALIZED)
3% Ni STEEL (38/45 TON)
1½-2% Ni STEEL (45/50 TON)
Cr ½ STEEL (45/50 TON)
3-5% Ni Cr STEEL (NORMAL IMPACT)
3-5% Ni Cr STEEL (LOW IMPACT)
Ni Cr Mo STEEL (75/85 TON)
SEMI-RANGE OF NOMINAL SHEAR STRESS DUE TO TORSION, q —TONS PER SQ. IN.
SEMI-RANGE OF NOMINAL DIRECT STRESS DUE TO BENDING, f —TONS PER SQ. IN.
D
LD
I

Fig. 7. *f/q* Curves
All for grooved specimens.

thus available, can quickly be determined using ordinary types of fatigue testing machines; a combined-stress fatigue testing machine is not necessary. Further, if doubt arises as to the applicability of the ellipse quadrant or ellipse arc for any particular material, the more conservative form—which will usually be the arc—should be used.

Theoretical and Actual Effect of Groove on Fatigue Strength. No exact analytical estimation appears to have been published of the theoretical stress concentration set up by the presence of

a small V-shaped circumferential groove in a shaft subjected to bending or torsion. Consideration of the work of Inglis[18] and Griffiths[19] indicated, however, that with the form and dimensions of notch used in the present tests, the following values of the theoretical stress concentration factor, K_t, are probably applicable without serious error:—

$$K_t \text{ (bending)} = 1 + 1.7\sqrt{\frac{d}{r}} \quad \ldots (3)$$

$$K_t \text{ (torsion)} = 1 + 0.6\sqrt{\frac{d}{r}} \quad \ldots (4)$$

where d is the depth of the notch and r is the root radius. As mentioned previously, a sharp V was never attained in the specimens; instead, a transition curve occurred at the root of the notch. The approximate radius of this curve was measured carefully on each specimen; fairly constant for each steel, it varied from steel to steel; the measured values for each steel have been used in equations (3) and (4) to calculate the values of K_t quoted below in Table 4; the value of d was constant throughout at 0.02 inch. The practical effect of the groove, which is supplied directly by the experimental data, is given by:—

K_f = fatigue strength reduction factor

$$= \frac{\text{fatigue strength of solid specimens}}{\text{fatigue strength of grooved specimens}}$$

A quantity which is often used to express the "sensitivity" of a material to a stress concentration is

Q = fatigue notch

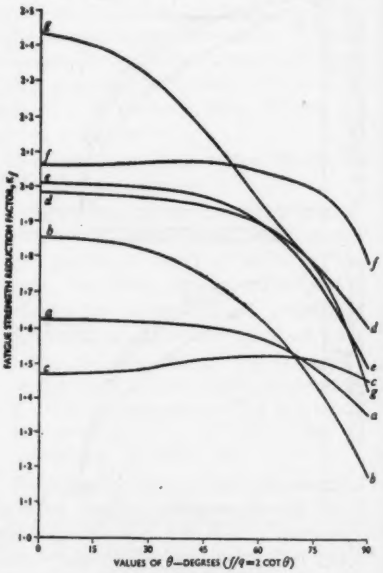
$$\text{sensitivity index} = \frac{K_f - 1}{K_t - 1}$$

the limiting values of Q being zero (when $K_f = 1$, the notch producing no reduction in the fatigue strength), and unity (when $K_f = K_t$ and the full theoretical effect is realized).

The comparison between approximate theory and experiment is made in the summarized data of Table 4, where the seven steels are arranged in ascending order of tensile strength.

For each material, K_t is considerably greater than K_f. Under reversed bending stresses, the value of the ratio K_t/K_f (column A of Table 4) falls steadily, if irregularly, with increase in tensile strength of the materials: a similar general trend is observed under reversed torsional stresses, although the results are more irregular. It will be noted that the very great difference in the notched-bar impact values of the same 3½ per cent nickel-chromium steel, tested in the normalized and embrittled conditions, is *not* reflected in their relative fatigue strengths, although a sharp notch is present: this fact is in accordance with previous results of similar tests on other materials.

The marked variation in sensitivity to the presence of the notch, which exists



$K_f = \dfrac{\text{fatigue limit of solid specimens}}{\text{fatigue limit of grooved specimens}}$.

	Material	U.T.S., tons per sq. in.
a	3 per cent nickel steel	34·1
b	0·4 per cent carbon steel (normalized)	42·0
c	3–3½ per cent nickel steel	46·8
d	Chromium-vanadium steel	48·7
e	3½ per cent nickel-chromium steel (normal impact)	53·0
f	3½ per cent nickel-chromium steel (low impact)	58·1
g	Nickel-chromium-molybdenum steel (75–80 tons)	80·5

Fig. 8. Fatigue Strength Reduction Factors due to 55 deg. V-groove

TABLE 6. COMPARISON OF THEORETICAL AND PRACTICAL EFFECT OF GROOVE ON FATIGUE STRENGTH

Material	Ultimate tensile strength, tons per sq. in.	Type* of stressing	Stress concentration factor, K_t (approx.)		Fatigue strength reduction factor, K_f		Ratios					Fatigue notch sensitivity index, Q	
			Bending $1+1{\cdot}7\sqrt{\frac{d}{r}}$	Torsion $1+0{\cdot}6\sqrt{\frac{d}{r}}$	Bending	Torsion	$\frac{K_t}{K_f}$ (A) Bending	$\frac{K_t}{K_f}$ (A) Torsion	$\frac{K_t \text{ (bending)}}{K_t \text{ (torsion)}}$ (B)	$\frac{K_f \text{ (bending)}}{K_f \text{ (torsion)}}$ (C)	$\frac{\text{(B)}}{\text{(C)}}$	Bending	Torsion
3 per cent nickel steel	34·1	B	18	—	1·63	—	11·0	—	2·57	1·20	2·14	0·04	0·06
		T	—	7	—	1·36	—	5·1					
0·4 per cent carbon steel (normalized)	42·0	B	18	—	1·85	—	9·7	—	2·57	1·57	1·64	0·05	0·03
		T	—	7	—	1·18	—	5·9					
3–3½ per cent nickel steel	46·8	B	13	—	1·47	—	8·8	—	2·36	1·01	2·34	0·04	0·10
		T	—	5·5	—	1·45	—	3·8					
Chromium - vanadium steel	48·7	B	13·8	—	1·99	—	6·9	—	2·89	1·24	2·33	0·08	0·16
		T	—	4·8	—	1·61	—	3·0					
3½ per cent nickel-chromium steel (normal impact)	58·0	B	9·5	—	2·01	—	4·7	—	2·53	1·35	1·87	0·12	0·18
		T	—	3·75	—	1·49	—	2·5					
3½ per cent nickel-chromium steel (low impact)	58·1	B	9·25	—	2·06	—	4·5	—	2·43	1·16	2·10	0·13	0·28
		T	—	3·8	—	1·78	—	2·1					
Nickel - chromium - molybdenum steel (75–80 tons)	80·5	B	7·2	—	2·43	—	3·0	—	1·89	1·71	1·11	0·23	0·15
		T	—	3·8	—	1·42	—	2·7					

K_t = theoretical stress concentration factor = ratio of theoretical stress at bottom of notch to nominal stress.

K_f = fatigue strength reduction factor = $\frac{\text{fatigue strength of solid specimens}}{\text{fatigue strength of grooved specimens}}$ (experimental values).

Q = fatigue notch sensitivity index = $\frac{K_f-1}{K_t-1}$.

* B denotes tests under reversed plane bending stresses ($\theta = 0$ deg.); T denotes tests under reversed torsional stresses ($\theta = 90$ deg.).

between the various materials, is brought out strikingly in Fig. 8, which shows, for each steel, the values of K_f* over the whole range of combined stresses from reversed bending to reversed torsional stresses. The family of seven curves vary between the two types exhibited by the nickel-chromium-molybdenum (75-80 tons) and the 3-3½ percent nickel (40-50 tons) steels. The former, which has the highest tensile strength of the group, is the most sensitive to reversed bending stresses but is less sensitive than others to reversed torsional stresses, while the latter material shows much the same degree of low sensitivity under all combinations of stress. No clue has yet been offered to the problem of these typical variations of sensitivity among various materials; they cannot be accounted for by single considerations of ultimate strength, ductility, chemical composition or form of microstructure; a metallic alloy is such a complex material that some complicated combination of a number of factors is probably involved. Until these are understood, a quantitative theory of fatigue is unlikely to be forthcoming; it is also most unlikely that the, necessarily, simple assumptions of the present theory of elasticity will prove to be applicable or helpful.

Influence of Tensile Strength, Composition, and Heat Treatment upon Fatigue Strength. The intrinsic strength under combined fatigue stresses, of each material, under the conditions of the present tests, is the fatigue strength developed when solid specimens are em-

* To avoid cumulative errors due to the calculation of K_f from uncorrected fatigue strengths of plain and notched specimens, the five values intermediate between b ($\theta = 0$ deg.) and t ($\theta = 90$ deg.) have been taken from the appropriate ellipse quadrant and ellipse arc, respectively: the numerator and denominator of K_f have also been used in terms of range of maximum shear stress, in order to avoid the difficulty of using two quantities, f and q.

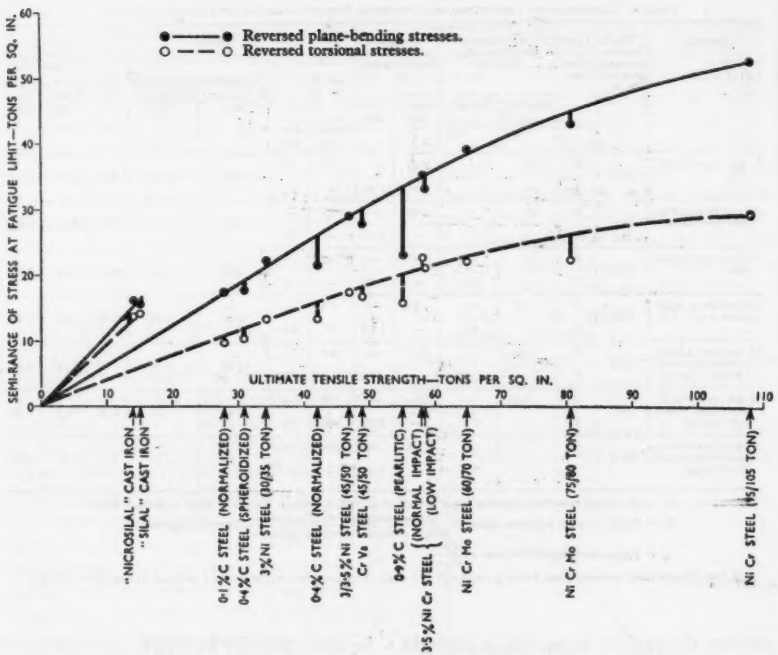

Reversed plane-bending stresses.
Reversed torsional stresses.
SEMI-RANGE OF STRESS AT FATIGUE LIMIT—TONS PER SQ. IN.
ULTIMATE TENSILE STRENGTH—TONS PER SQ. IN.
"NICROSILAL" CAST IRON
"SILAL" CAST IRON
0·1% C STEEL (NORMALIZED)
0·4% C STEEL (SPHEROIDIZED)
3% Ni STEEL (50/35 TON)
0·4% C STEEL (NORMALIZED)
1/3-1% Ni STEEL (45/50 TON)
Cr V₀ STEEL (45/50 TON)
0·9% C STEEL (PEARLITIC)
3½% Ni Cr STEEL (NORMAL IMPACT)
(LOW IMPACT)
Ni Cr Mo STEEL (60/70 TON)
Ni Cr Mo STEEL (75/80 TON)
Ni Cr STEEL (95/105 TON)

Fig. 9. Fatigue Strength in relation to Tensile Strength

ployed. Also, as it has been established that the fatigue strengths over the whole range of stress combinations explored are determined (by the ellipse quadrant or arc) once the "end-point" values are known, the latter only need be considered in relation to the effect of tensile strength, composition, and heat treatment, a considerable variety of which were purposely included in the program.

In Fig. 9, the fatigue ranges under reversed bending, and also reversed torsional, stresses of the ductile steels are plotted against ultimate tensile strength; smooth curves have been drawn to include the optimum fatigue values. The curve of reversed-bending stress/ultimate tensile strength is of the familiar type, the slope gradually falling as the ultimate tensile strength increases. All points on this curve represent very good fatigue resistance: even the value of the endurance ratio (fatigue limit/ultimate tensile strength) for the 108-ton steel is 48.6 percent, a very satisfactory figure. This curve, the characteristics of which are closely reproduced in the lower curve relating to torsional stresses, affords a convenient measure with which to examine the other variables involved. The 0.4 percent carbon steel, both in the normalized and spheroidized conditions, gives fatigue strengths which fall below the curve; neither structure has any clear relative advantages in this respect. The 0.9 percent carbon steel, tested in the pearlitic condition, develops very poor fatigue properties. The effect of embrittling the 3½ percent nickel-chromium steel produces a definite, but quite small, reduction in fatigue strength. In

the pair of steels chosen to have different compositions but practically the same tensile strength, the 3½ percent nickel steel is slightly superior to the chromium-vanadium steel.

Both cast irons develop uniquely high values of endurance ratio under each type of cyclic stress; for "Nicrosilal," the values were 1.15 and 0.97 for reversed bending and reversed torsion, respectively, the corresponding values for "Silal" being 1.05 and 0.95. The relative difference in brittleness—actually very small as judged by the notched-bar impact test—has had little, if any, effect on fatigue resistance.

Effect of Form of Specimen on Fatigue Resistance. Two materials—the 0.1 percent carbon steel (normalized) and the 3½ percent nickel-chromium steel (normal impact)—have each been tested, using both solid and hollow specimens, over the range of seven stress conditions. Each material, in each form of specimen, conforms closely to the ellipse quadrant relation. The "end-point" values were:—

0.1 percent carbon steel fatigue limits:

Reversed bending
± 17.4 tons per sq. in. (solid)

Reversed bending
± 17.5 tons per sq. in. (hollow)

Reversed torsion
± 9.85 tons per sq. in. (solid)

Reversed torsion
± 9.15 tons per sq. in. (hollow)

3½ percent nickel-chromium steel (normal impact) fatigue limits:

Reversed bending
± 35.0 tons per sq. in. (solid)

Reversed bending
± 33.6 tons per sq. in. (hollow)

Reversed torsion
± 22.8 tons per sq. in. (solid)

Reversed torsion
± 20.5 tons per sq. in. (hollow)

The effect of substituting the hollow for the solid specimen is thus to reduce the bending values by 0 and 4 percent, and the torsion values by 7 and 10 percent respectively. The reason for these differences—the modifications in the conditions of stress and strain caused by super-elastic straining within the safe range of stress—has been fully discussed elsewhere.[5, 20].

Having thus surveyed the scope and principal conclusions of the first stage of the research, we may turn to the second stage in which a start has been made on the exploration of the new field of combined fatigue stresses involving *four* independent variables.

STAGE 2. COMBINED FATIGUE STRESSES, EMPLOYING FOUR INDEPENDENT STRESS VARIABLES

In this second stage of the investigation it was required to impose on the test specimen two further independent variables, static bending stress and static torsional stress, in addition to the two cyclic variables, reversed bending and reversed torsional stresses, previously explored; using the previous nomenclature, the four required variables were $\pm f$, $\pm q$, f_s, and q_s. The design and construction of a suitable moderately high-speed fatigue testing machine had first to be accomplished and several different types were considered and carried to the stage of preliminary design; it was finally decided to develop a machine based in principle on the No. 1 type of machine whose proved performance, reliability, and constancy of calibration had been thoroughly satisfactory in every respect. A prototype was built that gave excellent performance under searching preliminary tests, and, with minor design modifications only, four machines were constructed, and put to work on the program.

The intrinsic fatigue properties of the material tested, under various combinations of all four stress variables, have been determined using solid specimens; to obtain data relating to the influence of some practical forms of discontinuity of section, it was decided, after consultation with the aircraft-engine industry, to carry out a similar program of tests on three series of specimens each containing one of the following: an oil hole, a small transition fillet, and a complete set of splines.

The Testing Machine. Time will permit a very brief description * of the operating principles only of this machine, which I will call "the No. 2 Combined Stress Fatigue Testing Machine." The method of generating the cyclic loading moments—by the centrifugal forces of unbalanced weights attached to revolving disks driven by a synchronous motor—is exactly similar to that employed in the No. 1 type of machine. Similarly, the setting angle, θ, between the vertical planes containing the loading arm and the longitudinal axis of the specimen, determines the ratio of the imposed cyclic bending and torsional moments. Also, as in the simpler machine, a suspensory spring system takes care of the inertia of the remaining vibrating and rotating parts. But, whereas in the No. 1 machine this was the sole function of the suspensory spring system, in the No. 2 machine initial deflections can be given independently to each pair of springs, which then impose on the specimen the required static bending and, or alternatively, static torsional moment. The principle of the loading system is illustrated diagrammatically in Fig. 10. The rear ends of each pair (right and left) of springs are held rigidly in a massive steel clamp which can slide, independently, vertically (and horizontally) in a casting attached to the base plate; these clamps can be locked in any desired relative position corresponding to deflections, d_1 and d_2, as shown in Fig. 10. The required values of d_1 and d_2 are pre-set very precisely, using measuring microscopes and an accurate static calibration. Next, after setting the specimen to the required θ value, the lengths of the side-arms shown in Fig. 10 are accurately adjusted so as to bring the disk-axle back to an exactly horizontal position † co-axial with fixed head-stock

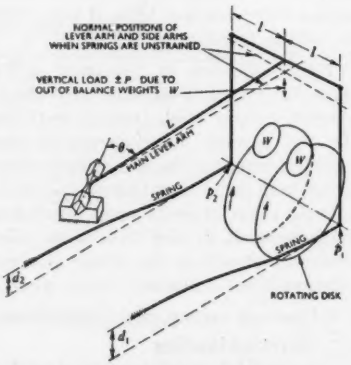


Fig. 10. Loading System of the No. 2 Combined Stress Fatigue Testing Machine

centers which provide a fixed datum line for the spring-loading system in relation to the baseplate of the machine. The machine is then ready for use; the static bending and, or alternatively, static torsional stresses applied to the specimen being determined by the spring settings d_1 and d_2, the ratio of f/q by θ and the actual values of f and q by the weights W and the (constant) speed of the revolving disk. Fig. 11 shows the general arrangement.

Test Material. This was a nickel-chromium-molybdenum-vanadium steel

* A detailed description of the machine and its calibration will be published elsewhere.[6]

† The final close adjustment of the axle height is effected by height and slip gauges from an accurately machined surface on the upper surface of the baseplate.

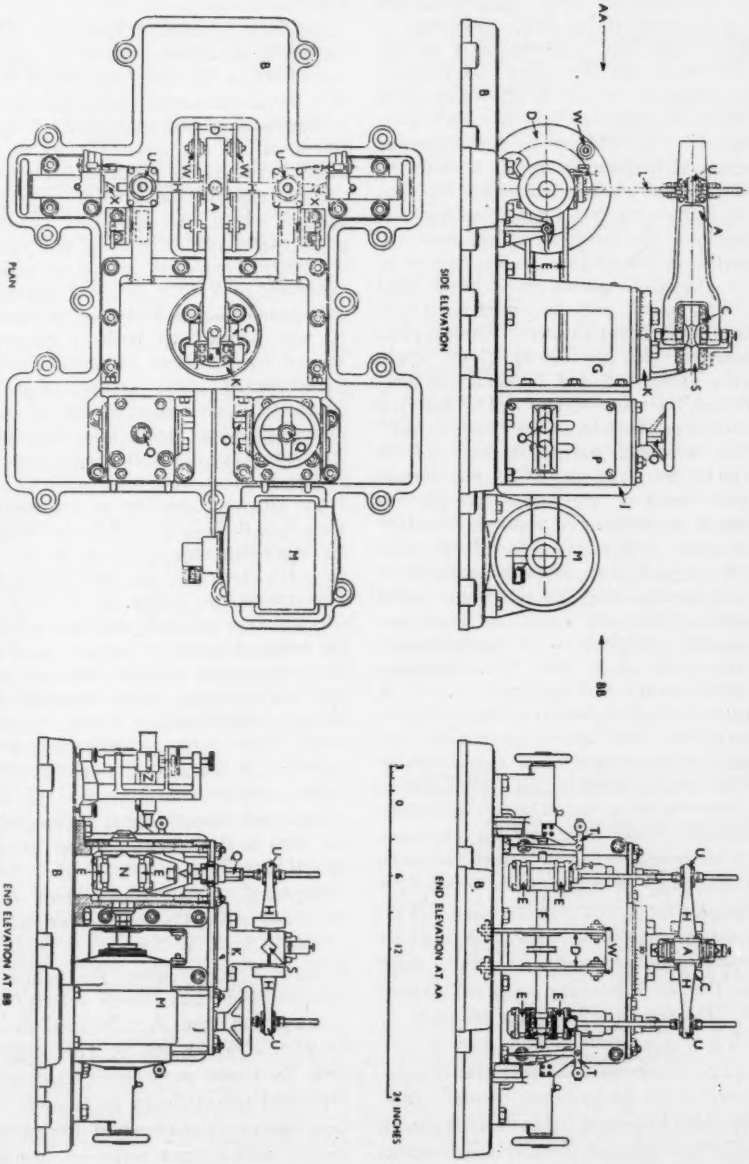


Fig. 11. General Arrangement of the No. 2 Combined Stress Fatigue Testing Machine

to B.S. Specification S65A for 65-ton nickel-chromium steel. To provide for future tests on full-sized specimens, a special 32½-ton cast was made of this alloy steel. An interesting metallurgical investigation of one of the forty-seven ingots forming the cast has been published.[21] In addition to the extensive series of tests made on this steel in the No. 2 type machine, a series was put through on the No. 1 type machine; the results of the latter have been reported earlier in this address under the title of the "nickel-chromium-molybdenum steel (60-70 tons)" and the summarized particulars of heat treatment, chemical composition, etc., previously given, apply (see Tables 1, 2, and 3). To avoid confusion, the test material will be referred to in what follows as the "S65A steel." The recorded tensile strength of 65 (64.8) tons per sq. in. is the normal value for this steel, which possess excellent ductility (67 percent reduction of area) and notched-bar Izod value (90 ft.-lb.) The material exhibits a well-marked drop at the upper yield point of 61.3 tons per sq. in. Final machining operations on all the test specimens were made after the rough-machined blanks had been heat treated. A batch method of heat treatment was unavoidable, but every precaution was taken to preserve constant conditions for each batch; nevertheless, small differences within a batch were sometimes revealed, under the searching conditions of the fatigue tests, by "scatter" on some of the S/N diagrams. The amount of this scatter is not excessive for a heat-treated, high-alloy steel, for, of the seventy-one fatigue determinations made on the No. 2 machine, sixty are classed as "D", nine as "LD", and two only as "I".

Test Program. The program of combined stress fatigue tests divides naturally into two parts; that carried out on solid specimens of circular cross-section and that carried out on each of three selected forms of discontinuity of section.

(a) Tests on Solid Specimens. The form and dimensions of the specimens are shown in Fig. 3*d*. The fatigue limit of a metal under cyclic reversed stresses is dependent on the superimposed static stress, which can have any value between zero and the ultimate strength of the material. In the present investigation in which two kinds of superimposed static stress, bending and shear, were to be explored, it was, therefore, necessary to select a series of values of each kind of stress (bending or shear). It was decided to build a program around three values of each kind of superimposed static stress; also, to couple with each combination of these three values not more than five combinations (i.e. five values of θ) of reversed bending and reversed torsional cyclic stresses; the five values selected were $\theta = 0, 29.7, 53.1, 75.9$, and 90 deg., corresponding to $\pm f/\pm q = \infty$ ($q = 0$), 3½, 1½, ½, and 0 ($f = 0$). The maximum values of f_s and q_s selected were one-half the fatigue limits for repeated cycles of stress (in which the stress varied cyclically between zero and a maximum) under bending and torsion respectively. These repeated stress limits were determined experimentally in the No. 2 machine: their values were:—

Repeated stress fatigue limit (bending) = 0–69 tons per sq. in., i.e. 34½ ± 34½.

Repeated stress fatigue limit (torsion) = 0–44½ tons per sq. in., i.e. 22¼ ± 22¼.

Based on these results, the values of superimposed static stress selected for investigation were: $f_s = 34½, 17¼$, and 0; $q_s = 22¼$, 11, and 0. The program built up round these static stresses is expressed schematically in Table 5. The nine squares, numbered in Roman numerals, each cover a series of combined stress fatigue tests in which the super-

TABLE 5. SCHEMATIC REPRESENTATION OF THE TEST PROGRAMME CARRIED OUT ON SOLID SPECIMENS (S65A)

Superimposed static torsional stress, tons per sq. in.	Superimposed static bending stress, tons per sq. in. $f_s = 0$	$f_s = 17\frac{1}{4}$	$f_s = 34\frac{1}{2}$
$q_s = 0$	(1) $f/q = \infty$ ($q = 0$) (19) $f/q = 3\frac{1}{4}$ (20) $f/q = 1\frac{1}{2}$ (21) $f/q = \frac{1}{2}$ (4) $f/q = 0$ ($J = 0$) I	(2) $f/q = \infty$ (28) $f/q = 1\frac{1}{2}$ (13) $f/q = 0$ II	(3) $f/q = \infty$ (14) $f/q = 0$ III
$q_s = 11$	(7) $f/q = \infty$ (29) $f/q = 1\frac{1}{2}$ (5) $f/q = 0$ IV	(9) $f/q = \infty$ (22) $f/q = 3\frac{1}{4}$ (23) $f/q = 1\frac{1}{2}$ (24) $f/q = \frac{1}{2}$ (15) $f/q = 0$ V	(11) $f/q = \infty$ (16) $f/q = 0$ VI
$q_s = 22\frac{1}{2}$	(8) $f/q = \infty$ (6) $f/q = 0$ VII	(10) $f/q = \infty$ (17) $f/q = 0$ VIII	(12) $f/q = \infty$ (25) $f/q = 3\frac{1}{4}$ (26) $f/q = 1\frac{1}{2}$ (27) $f/q = \frac{1}{2}$ (18) $f/q = 0$ IX

imposed static stresses, given by the headings of the vertical and horizontal columns in which the particular square is situated, are held constant. In each square, the fatigue limits to be determined are sufficiently specified by a statement of the f/q ratio; each test is numbered for reference purposes. The importance of the stress combinations represented by squares I, V, and IX merited the full series of five values of reversed bending stresses/reversed torsional stresses: in the remaining cases, considerations of economy in the testing time and numbers of specimens led to smaller numbers, three for squares II and IV, two for squares III, VI, VII, and VIII. Each of the twenty-nine separate fatigue limits involved was determined, on one of the No. 2 type machines, by the usual method of endurance testing on a minimum basis of 10^7 cycles. As with the No. 1 type machine, a trip gear stopped the machine when a sensible increase occurred in the cyclic range of strain of the specimen; all tests were thus stopped in the very early stage of fatigue cracking.

(b) Tests on Specimens Containing Discontinuities of Section. The three types* of specimen are:—

†Hollow specimen having a drilled radial oil hole, to Fig. 3*e*.

Hollow specimen having a transition fillet of small radius, to Fig. 3*f*.

Hollow specimen having six deep splines, based on British Standard Specification, No. 46, Part 2, 1929, Table 14, U.K. Fit., to Fig. 3*g*.

On each type, the test program shown in Table 6 was carried out. After the explanation of the previous table, Table 6 will be self-explanatory. It was considered unnecessary, from practical design considerations, to explore values of f_s and q_s greater than 17¼ and 11 tons per sq. in. respectively. In these series of tests, except where otherwise stated, all values of f, q, f_s, and q_s, refer to nominal stress values, calculated in the usual way without reference to the stress concentration effect produced by the discontinuity: that is discussed separately.

TABLE 6. SCHEMATIC REPRESENTATION OF THE TEST PROGRAMME CARRIED OUT ON HOLLOW SPECIMENS CONTAINING ONE OF THREE FORMS OF DISCONTINUITY OF SECTION

Superimposed static torsional stress, tons per sq. in.	Superimposed static bending stress, tons per sq. in. $f_s = 0$	$f_s = 17\frac{1}{4}$
$q_s = 0$	(1) $f/q = \infty$ ($q = 0$) (13) $f/q = 3\frac{1}{4}$ (9) $f/q = 1\frac{1}{2}$ (14) $f/q = \frac{1}{2}$ (5) $f/q = 0$ ($J = 0$) I	(2) $f/q = \infty$ (10) $f/q = 1\frac{1}{2}$ (6) $f/q = 0$ II
$q_s = 11$	(3) $f/q = \infty$ (11) $f/q = 1\frac{1}{2}$ (7) $f/q = 0$ III	(4) $f/q = \infty$ (12) $f/q = 1\frac{1}{2}$ (8) $f/q = 0$ IV

*A hollow form of specimen was adopted as representing a nearer approach to a practical engine component in which such discontinuities are present. The results may not be strictly comparable with the intrinsic fatigue strength of the material as exhibited by solid specimens, but these hollow specimens are "thick-walled" and the "form" effect is probably small. it is certainly very small in comparison with the reduction in strength due to the discontinuity.

† When under test, all specimens containing the oil-hole were mounted in the fatigue testing machine so that the axis of the hole lay exactly in the vertical plane of the applied bending moment. The splined-shaft specimens were all mounted so that this plane passed through the centers of an opposite pair of spline crests, although preliminary fatigue tests had, in fact, shown that the same fatigue limit was obtained with the specimens mounted in the alternative position in which the vertical plane of the applied bending movement passed through the centers of an opposite pair of spline troughs.

RESULTS AND CONCLUSIONS

Solid Specimens. Each individual test is identified by the quoted reference number from Table 5.

Commencing with the influence of superimposed static stress, of a similar and dissimilar type (bending and torsion) singly and in combination, on the fatigue limits under simple reversed bending and torsional stresses, the results merit detailed examination as they determine the two-dimensional framework on which this field of combined stresses will be erected; for this purpose, Fig. 13 has been prepared. Fig. 13*a* shows the effect on the fatigue limit under simple reversed bending stresses: Fig. 13*b* shows the effect on the fatigue limit under simple reversed torsional stresses.

Consider first the influence of similar types of stress. In Fig. 13*a*, the reversed bending fatigue limit is $\pm$ 37.8 tons per sq. in. (test 1); superimposing static bending stresses only of 17¼ and 34½ tons per sq. in., reduces the fatigue limit to values of $\pm$ 35.8 (test 2) and $\pm$ 34.5 tons per sq. in. (test 3), a maximum reduction of 8.7 percent only. In Fig. 13*b*, the reversed torsional fatigue limit is $\pm$ 24.0 tons per sq. in. (test 4); superimposing static torsional stresses of 11 and 22¼ tons per sq. in., reduces the fatigue limit to $\pm$ 22.0 (test 5) and $\pm$ 22.2 tons per sq. in. (test 6), a maximum reduction of 8.3 percent only. Thus the effect of superimposing quite high values of static stresses of the same kind is definite but quite small, and this is an important characteristic from the general design aspect. It would be idle to attempt to express in terms of stress only the shape of these M/R curves for this material, for it is well established that the M/R relation varies both with the material under test and the type of applied stressing. With increasing values of M, the value of R diminishes, in general much more rapidly for

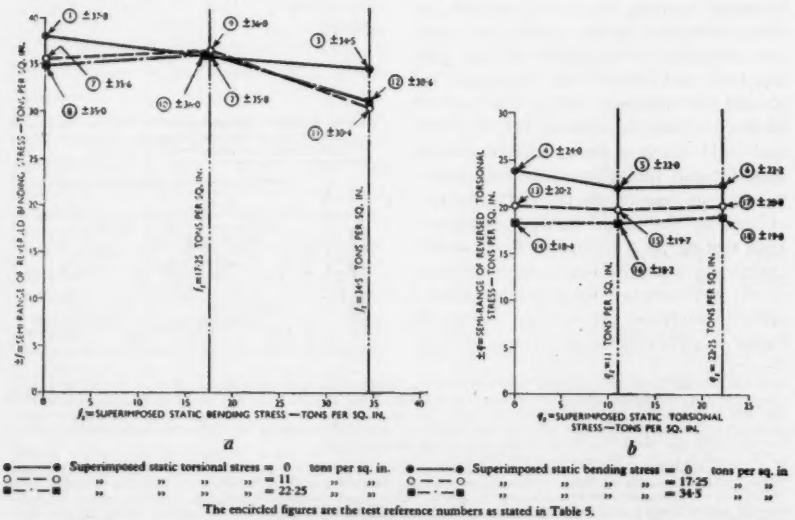
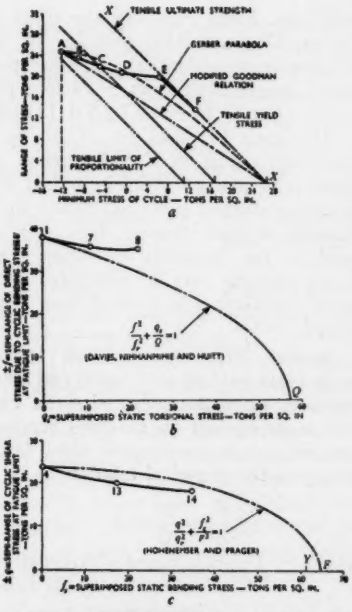

a
b
Superimposed static torsional stress = 0 tons per sq. in.
" " = 11 " "
" " = 22·25 " "
Superimposed static bending stress = 0 tons per sq. in
" " = 17·25 " "
" " = 34·5 " "
The encircled figures are the test reference numbers as stated in Table 5.

Fig. 13. Effects of Superimposed Static Stresses, Singly and in Combination, on the Fatigue Limits under Reversed Cyclic Stresses. Solid Specimens, S65A Steel

soft, ductile non-ferrous metals than with steels, and more sharply under direct stresses than with flexural or torsional stresses for most metals and alloys; the shape of the static load-elongation diagram to fracture is also a factor. Nevertheless, as far as engineering steels are concerned, the effect of mean stress is much less than is often assumed. As an extreme example, a series of tests[22] made on a dead-mild steel under cyclic direct stresses may be recalled: the results are summarized in Fig. 14*a*. Under the extensive series of stress ranges which yielded the six fatigue limits represented by the points A, B, C, D, E, and F, the elastic limit and yield point of the material were exceeded and the ultimate tensile strength very closely approached; very considerable deformations of the specimens occurred. Yet, until the maximum stress of the cycle was nearly equal to the ultimate tensile strength, the drop in fatigue strength, although quite definite, was surprisingly small; neither the "Gerber parabola" nor the linear "modified Goodman relation" predicts the fatigue resistance of this material. In fact, one would not expect any such relations to have a general application even to the group of engineering steels: the result of a study[22, 52] of deformation characteristics of metals, using X-ray precision methods, leads to the conclusion that the total damage suffered is probably directly attributable to the combined effects of the applied range of stress and the damage caused to the crystalline structure by the deformation due to the maximum stress of the cycle; the latter factor involves the stress-strain deformation characteristics of the material. Thus, viewed against this background, it is not surprising to find that the present S65A steel, with its high ratio of yield/ultimate tensile strength* and especially when tested under flexure and torsion, shows relatively small


TENSILE ULTIMATE STRENGTH
GERBER PARABOLA
MODIFIED GOODMAN RELATION
TENSILE YIELD STRESS
TENSILE LIMIT OF PROPORTIONALITY
MINIMUM STRESS OF CYCLE — TONS PER SQ. IN.
a
(DAVIES, NIHMANNIMME AND HUFF)
Q=SUPERIMPOSED STATIC TORSIONAL STRESS — TONS PER SQ. IN
b
(HOHENEMSER AND PRAGER)
f=SUPERIMPOSED STATIC BENDING STRESS — TONS PER SQ. IN.
c

a Fatigue resistance of normalized mild steel under cyclic direct stresses (Gough and Wood).
b Effect of static torsional stress on fatigue limit under reversed bending stresses (S65A steel).
Q = torsional modulus of rupture = 57·3 tons per sq. in.
c Effect of static bending stress on fatigue limit under reversed torsional stresses (S65A steel).
F = ultimate tensile strength = 64·8 tons per sq. in.
Y = yield point in tension = 61·3 tons per sq. in.

Fig. 14. Effect of Mean Stress (Mild Steel and S65A Steel) and Comparison with Certain Criteria

drops in fatigue resistance under even very high values of applied static stresses of similar kinds.

In Fig. 13, the influence of dissimilar types of static stress, also of combinations of similar and dissimilar types, merit detailed scrutiny. Tests 1, 7, and 8 on Fig. 13*a* show the effect of imposing static torsional stresses on the reversed bending fatigue limit; adding 11 and 22¼ tons per sq. in. reduces the fatigue limit from ± 37.8 tons per sq. in. (test 1) to ± 35.6 (test 7) and ± 35.0 (test 8) tons per sq. in. respectively, a maximum reduction of 7.4 per-

* The values of the upper yield point and tensile strength of this material were, respectively. 61.3 and 64.8 tons per sq. in.

cent only. From Fig. 13*b*, the effect of adding 17¼ and 34½ tons per sq. in. bending stress reduces the reversed torsional fatigue limit from ± 24 tons per sq. in. (test 4) to ± 20.2 (test 13) and ± 18.4 (test 14) tons per sq. in., or a maximum reduction of 23.4 percent, an appreciable amount which is, in fact, the greatest reduction produced by any combination of superimposed static stresses used in the investigation. These two series of tests may be compared with certain criteria which have been suggested.

Davies, Nimhanmimie, and Huitt† made some tests on mild steels and alloy steels to study the effect of static torsion superimposed on reversed bending stresses and obtained results which fitted closely to the empirical relation

$$\frac{f^2}{f_r^2}+\frac{q_s}{Q}=1$$

where $\pm f$ = safe range of direct stress due to the cyclic bending stresses;

q_s = corresponding superimposed static torsional stress;

$\pm f_r$ = fatigue range under reversed bending stresses only; and

Q = ultimate static torsional strength.

Fig. 14*b* shows the above expression plotted in relation to the strength properties of the S65A steel, together with the actual results of tests 1, 7, and 8: these results do not conform to the proposed relation. Hohenemser and Prager[23] investigated the effect of static tensile stress superimposed on alternating torsional stresses, using a mild steel. They found their results conformed to the ellipse quadrant relation

$$\frac{q^2}{q_r^2}+\frac{f_s^2}{F^2}=1$$

where $\pm q$ = fatigue limit of cyclic shear stress;

f_s = superimposed static tensile stress;

F = ultimate tensile strength; and

$\pm q_r$ = fatigue range under reversed shear stresses only.

The conditions of the present test are not quite comparable, as static bending stresses have been employed in place of the static direct stresses used by Hohenemser and Prager. As a matter of interest, however, an ellipse quadrant has been drawn in Fig. 14*c*, drawn through the point F denoting tensile ultimate strength, as the bending ultimate strength of S65A is not determinate. The results of tests 4, 13, and 14 are shown on the diagram but do not conform to the quadrant relation shown.

The remaining test data plotted on Fig. 13 relate to combinations of superimposed static stresses of dissimilar kinds. In Fig. 13*a*, tests 9 and 10 show that adding torsional stresses of either 11 or 22¼ tons per sq. in. to a previously applied bending stress of 17¼ tons per sq. in. produces no appreciable additional reduction in the value of the fatigue limit under reversed bending stresses. A greater effect is produced when either of these same torsional stresses are superimposed on a static bending stress of 34½ tons per sq. in., as shown by tests 11 and 12; the most severe stress conditions of test 12, i.e., 34½ tons per sq. in. bending allied with 22¼ tons per sq. in. torsion, reduces the original reversed bending fatigue limit from ± 37.8 (test 1) to ± 30.6 tons per sq. in., a total reduction of 19.1 percent only. Fig. 13*b* shows the same general effect on the reversed torsional fatigue limit, as disclosed by the results of tests 15, 16, 17, and 18. The most severe stress conditions of test 18, i.e. 34½ tons per sq. in. bending allied with 22¼ tons per sq. in. torsion, reduce the original reversed torsional fatigue limit

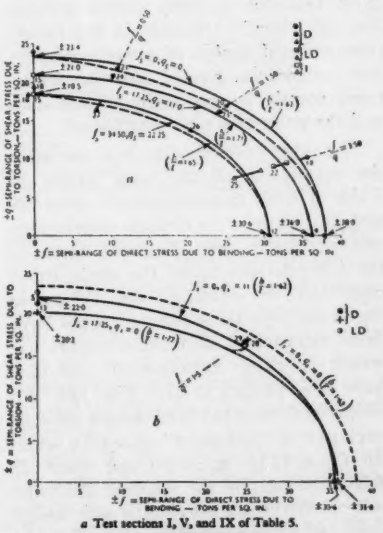
† See discussion (p. 66) of paper by Gough and Pollard.[4]

from ± 24 (test 4) to ± 19 tons per sq. in.. a reduction of 20.8 percent.

Reviewing this complete series of results, one finds the data are satisfactorily regular and consistent and can reliably be used, by interpolation, to deduce the safe range of reversed bending or reversed torsional stress under any required combination of superimposed static stresses within the wide limits investigated. That part of the combined stress field has, therefore, been satisfactorily surveyed for design purposes in regard to the S65A steel; the same general trend of results would probably be exhibited by other high yield-ratio alloy steels of this type. The effect of the addition of one static stress, or a combination, is quite definite and regular, but the amount of the reduction, caused even by static stresses which are high in relation to the ultimate strengths of the material, is relatively small. The effect of superimposing static bending stresses on cyclic torsional stresses is more damaging than that of static torsional stresses imposed on cyclic bending stresses.

We can now consider the results of those series of tests in which the values of the static stresses are held constant and the fatigue strength is determined under various combinations of cyclic bending and torsional stresses.

There are three principal series in each of which the fatigue ranges have been determined for five values of f/q; these are shown in squares I, V, and IX of Table 5, the superimposed static stress system held constant during each series being: $f_s = 0$, $q_s = 0$ (series I); $f_s = 17\frac{1}{4}$, $q_s = 11$ (series V); $f_s = 34\frac{1}{2}$, $q_s = 22\frac{1}{4}$ (series IX). The cyclic bending and shearing stress components of these at the fatigue limits are plotted in Fig. 15*a*. Using the method of least squares, the best fitting ellipse quadrant has been drawn for each series of data; the corresponding ellipse arcs are also shown for series I and IX. In series I, the data conform closely to the ellipse quadrant curve; the quadrant and the arc are not widely separated, but the quadrant provides the better fit. The results of this series, in which the mean stress of each kind of stress cycle has zero value, thus provide a further confirmation of the previous general investigation of stage I, i.e. that the f/q relation of ductile steels, when tested in the form of specimens free from discontinuities of section, is represented by an ellipse quadrant. It will be seen from Fig. 15*a* that the results of series V, when static stresses ($f_s = 17\frac{1}{4}$, $q_s = 11$) are in operation, is also closely represented by the corresponding ellipse quadrant: it is unnecessary to show the corresponding ellipse arc as this obviously cannot provide so good a fit to the test data as the quadrant. The data of series IX ($f_s = 34\frac{1}{2}$, $q_s = 22\frac{1}{4}$) are somewhat irregular but, within the accuracy of these data, are satisfactorily represented by the quadrant; in this case



a Test sections I, V, and IX of Table 5.
——— Ellipse quadrants.
– – – – Ellipse arcs.
b Test sections II and IV of Table 5.
——— Ellipse quadrants.
– – – – Curve reproduced from Fig. 15*a*, for comparison.

Fig. 15. Effect of Superimposed Static Stress on the f/q Relation. S65A Steel, Solid Specimens

the arc and quadrant are not widely divergent and either curve could, without serious error, be said to fit the irregular data; but, in view of the weight of evidence now obtained that solid specimens of the ductile steels conform much more closely to the quadrant relation, it is justifiable to assume that, in this case also, the quadrant relation holds.

Thus, a further and most important conclusion emerges from these tests. The effect of superimposing up to very considerable values of static bending and torsional stress merely affects the actual values of the "end-points" b and t; the f/q relation remains the ellipse quadrant: when these "end-point" values have been determined experimentally for any required values of f_s and q_s, the fatigue limit for any required ratio f/q is simply calculated, a fact which should be of considerable value for design purposes. As reliable data regarding the M/R relations of many metals and alloys are already available in the literature, rational design of many engineering components should now be facilitated merely by use of published data and the principles now established.

The above discussion has included the results of all the tests shown in Table 5 with two exceptions—tests 28 and 29. Clearly, no definite conclusions can be drawn regarding the form of the f/q relations under the static stress conditions of series II and IV as, in each case, only three fatigue ranges have been determined. But these results are worth a brief examination and they have been plotted to give Fig. 15*b*. An ellipse quadrant has been drawn through each pair of "end-point" values. In series II ($f_s = 17\frac{1}{4}$, $q_s = 0$), test point 28 falls well outside the relevant quadrant, which, therefore, provides a safe design basis, while in series IV ($f_s = 0$, $q_s = 11$) test point 29 is very close to its predicted curve. It can certainly be said that, making allowance for small variations in the physical properties of the steel from batch to batch, there is nothing in these isolated tests to throw serious doubt on the general applicability of the ellipse quadrant to any values of f_s and q_s within the wide limits investigated.

Specimens Containing Discontinuities of Section. Each individual test will be identified by the reference number in Table 6.

(a) Effect of Superimposed Static Stresses, Singly and in Combination on the Fatigue Limits under Reversed Cyclic Stresses. The data relating to the tests made on specimens containing a radial hole are plotted in Fig. 16*a* and *b*.

In Fig. 16*a*, the fatigue limit under simple reversed bending stresses is $\pm$ 16.8 tons per sq. in. (test 1): this value is slightly reduced to $\pm$ 15.4 tons per sq. in. (test 2), i.e. by 8.3 percent, by superimposing a static bending stress of 17¼ tons per sq. in. The further addition of a static torsional stress of 11 tons per sq. in. (tests 3 and 4) to each of these conditions, produces no appreciable further reduction. In Fig. 16*b*, the fatigue limit under reversed torsional stresses is $\pm$ 11.7 tons per sq. in.: imposing a static torsional stress of 11 tons per sq. in. reduces this value, by 3.4 percent only, to $\pm$ 11.3 tons per sq. in. (test 7): the further addition of 17¼ tons per sq. in. static bending stress has no appreciable effect, the recorded fatigue limit being $\pm$ 11.2 tons per sq. in. (test 8). The effect of adding a static bending stress of 17¼ tons per sq. in. only reduces the simple reversed torsional limit from $\pm$ 11.7 tons per sq. in. (test 5) to $\pm$ 11.0 tons per sq. in. (test 6), i.e., by 6 percent.

The data relating to the fillet specimens have similarly been plotted in Fig. 16*c* and *d*. As shown in Fig. 16*c*, the fatigue limit under simple reversed bending stresses is $\pm$ 22.5 tons per sq. in. (test 1) and superimposing a static bending stress of 17¼ tons per sq. in. (test 2) has no effect. The further addition of a static shearing stress of 11

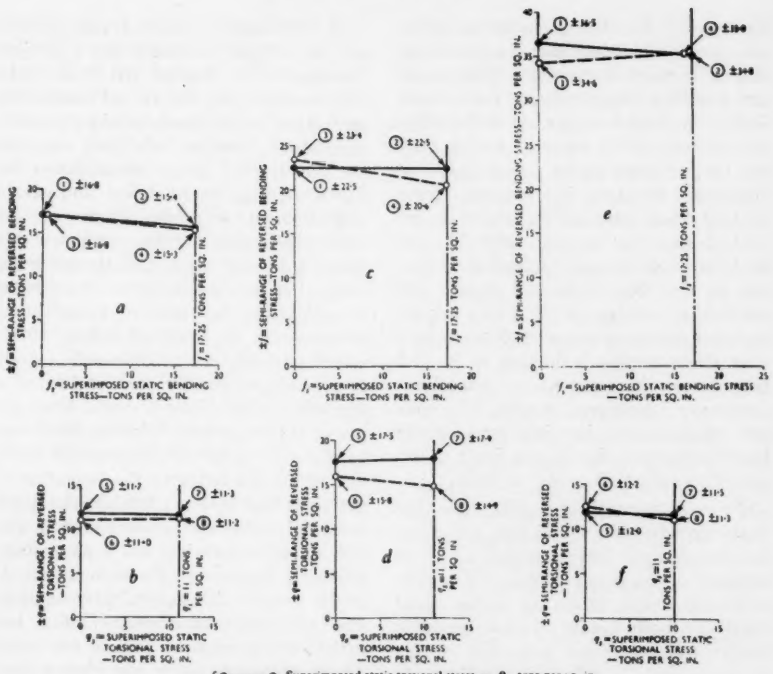

a
b
c
d
e
f
f_s = SUPERIMPOSED STATIC BENDING STRESS—TONS PER SQ. IN.
f_t = SUPERIMPOSED STATIC TORSIONAL STRESS —TONS PER SQ. IN.
f_s = 17·25 TONS PER SQ. IN.
f_t = 11 TONS PER SQ. IN.
a, c, e Superimposed static torsional stress = 0 tons per sq. in.
" " = 11 " "
b, d, f Superimposed static bending stress = 0 tons per sq. in.
" " = 17·25 " "
a and b Specimens with oil hole.
c and d Specimens with small fillet.
e and f Specimens with splined shafts.
The encircled figures are the test reference numbers as stated in Table 6.

Fig. 16. Effects of Superimposed Static Stresses, Singly and in Combination, on the Fatigue Limits under Reversed Cyclic Stresses (S65A Steel)

tons per sq. in. reduces the fatigue limit to ± 20.6 tons per sq. in. (test 4), or a total reduction of 8½ percent. Superimposing a static torsional stress only of 11 tons per sq. in. apparently slightly raises the fatigue limit to ± 23.4 tons per sq. in. (test 3) but this result must be due to slight differences in the material and it can be assumed that this static stress has, in fact, no effect. In Fig. 16*d*, the fatigue limit under simple reversed torsional stresses is ± 17.5 tons per sq. in. (test 5): superimposing a static shear stress of 11 tons per sq. in. apparently raises the fatigue limit slightly to ± 17.9 tons per sq. in. (test 7) which can also be ignored as a real effect. A static bending stress of 17¼ tons per sq. in. reduces the fatigue limit to ± 15.8 tons per sq. in. (test 6), while the combined effect of imposing 17¼ tons per sq. in. static bending and 11 tons per sq. in. static torsional stress reduces the fatigue limit to ± 14.9 tons per sq. in. (test 8)—14.9 percent total reduction.

Lastly, in Fig. 16*e* and *f* are plotted the data relating to the splined shaft specimens. The fatigue limit under simple reversed bending stresses, ± 36.5 tons per sq. in. (test 1) is slightly reduced to ± 34.8 tons per sq. in. (test 2), i.e., by 4.7 percent, by superimposing 17¼ tons per sq. in. in static bend-

ing stress; the further addition of 11 tons per sq. in. static shear produces no further reduction (test 4). This shear stress acting alone reduces the fatigue limit to ± 34.6 tons per sq. in. (test 3), a reduction of 5.2 percent. In Fig. 16*f*, the fatigue limit under simple reversed shearing stresses is reduced from ± 12.0 tons per sq. in. (test 5) to ± 11.5 tons per sq. in. (test 7), i.e., by 4.2 percent, by superimposing 11 tons per sq. in. static torsional stress, and the further addition of 17¼ tons per sq. in. static bending stress produces only a very slight further reduction to ± 11.3 tons per sq. in. (test 8), or a total reduction of 5.8 percent. Adding 17¼ tons per sq. in. static bending stress alone has no effect on the fatigue limit under simple reversed shearing stresses.

To summarize, these results show that in the presence of a radial hole, the effect on the fatigue limits under reversed bending or torsional stresses of superimposing static stress of either kind singly or both kinds simultaneously is small; the maximum reduction in fatigue strength is 8.3 percent. When a small transition fillet is present, the effect of the superimposition is more definite, particularly on the reversed torsional fatigue limit, which suffered a maximum reduction of 14.9 percent. The effect of superimposing the same values of static stresses on the model splined shafts was extremely small, the maximum reduction in fatigue strength being 5.8 percent. It will be seen, however, that these same forms of discontinuity are very sensitive to cyclic stresses and this entirely different response to static and cyclic stressing is of the greatest significance to design; the majority of fatigue failures in practice occur at such discontinuities of contour or cross-section.

It is convenient first to discuss the stress-concentration effects caused by the three forms of discontinuity before examining the form of f/q relation disclosed by the test data.

(b) Theoretical and Actual Effects of the Discontinuities on Fatigue Strength. (1) Radial Oil Hole. The stress system and theoretical stress concentration factor produced by the presence of a cylindrical hole have attracted the attention of many investigators using analytical, photo-elastic and strain measurement methods. Many of the cases investigated relate only to thin plates subjected to simple stressing actions, but a consideration of selected investigations has enabled approximate estimates to be made of values of K_t which probably relate sufficiently closely to the present stress system and form of specimen as to afford a useful comparison of theory and experiment. As shown by Kirsch,[24] under uniform tensile stress and at the boundary of the hole, two of the principal stresses vanish, the third principal stress—a circumferential tensile stress—operating on a plane containing a diameter of the oil-hole. Cox,[17] in the course of a general investigation into the effect of elliptical holes, has extended this case to that of combined direct stress and shear and shown that, when these stresses become reversed cyclic stresses, the analysis reveals that the limiting condition for failure is such that the f/q relation should be an ellipse arc. Further, if fatigue failure occurs on the plane subjected to the greatest range of principal stress, then, by the analyses of Kirsch and Cox, the fatigue cracks should be initiated at the boundary of the oil-hole on a plane making an angle of $\frac{1}{2}\theta$ with a transverse section of the specimen. In the present tests the inclination of the trace of every fatigue crack at its point of initiation was measured and the agreement with the value of $\frac{1}{2}\theta$ was so close that there is little doubt that the plane of fracture is as indicated by theory.

The work of Kirsch and Cox relate to the assumed conditions of a hole having a diameter infinitesimally small in relation to the specimen size; other investigators have shown that the value of

the theoretical stress-concentration factor, K_t, is dependent on the ratio of the diameter of the hole, d, to that of the specimen, D. The actual values of K_t given by the analyses of Kirsch and Cox are not, therefore, directly applicable, but consideration of the work of Frocht,[25, 26] Wahl and Beeuwkes,[27] Peterson and Wahl[12], Thum and Kirmser,[28] indicates that the following approximate values apply to the present specimen,

where $\frac{d}{D} = \frac{1}{10}$:—

K_t (reversed bending stresses) = about 2.5–2.7, say, 2.6.
K_t (reversed torsional stresses) = about 2.0.

In the present specimens, the intersection of the oil-hole with the outside surface of the shaft is rounded off with a small radius, but this radius would not be expected to have any appreciable strengthening effect: its value in practice is to remove machining burrs, as these would cause a definite reduction in fatigue strength. For comparison of K_t with the actual reduction in fatigue strength caused by the presence of the oil-hole, the experimental values of the Fatigue Strength Reduction Factor, K_f, are listed in Table 7, where the figures in brackets refer to the test reference numbers of Table 6. As K_f is obtained by dividing two experimentally determined quantities (fatigue strength of solid specimens/fatigue strength of specimens with oil-hole) each of which is subject to errors arising from small variations in material, which may be cumulative, large variations in K_f might occur. In spite of this, the values for each value of θ are remarkably constant, leading to two main conclusions:

That the average value of K_f for reversed bending stresses is about 2.25 and 1.95 for reversed torsional stresses, with a fairly regular change between the end values. Comparison with the approximate values of K_t (2.6 for bending, 2 for torsion) shows that the presence of the oil-hole produces, in this steel, a reduction in fatigue strength such that $K_f/K_t = 86\frac{1}{2}$ and $97\frac{1}{2}$ percent respectively; the theoretical effect is largely realized.

That the value of K_f is determined primarily by the applied cyclic stresses and is not appreciably affected by the superimposed static stresses—a very important conclusion, suggested perhaps by general knowledge of fatigue characteristics.

(2) Small Transition Fillet. Nearly all the specimens failed similarly, by a transverse crack situated at the junction of the fillet with the parallel central portion of the test-piece or slightly removed from that junction and within the fillet.

By strain-measurements, photo-elastic methods, and the use of the electrical analogy, the work of Peterson and Wahl,[12] Frocht,[25] and Jacobsen[29] indicates the following approximate values of the stress-concentration factor for

TABLE 7. FATIGUE STRENGTH REDUCTION FACTORS, K_f: SPECIMEN WITH OIL-HOLE

Superimposed static stress, tons per sq. in.		Values of K_f (uncorrected)				
Bending, f_s	Torsion, q_s	$\theta = 0$	$\theta = 29{\cdot}7$	$\theta = 53{\cdot}1$	$\theta = 75{\cdot}9$	$\theta = 90$
0	0	2·25 (1)	2·49 (13)	2·23 (9)	1·95 (14)	2·05 (5)
17½	0	2·32 (2)	—	2·42 (10)	—	1·84 (6)
0	11	2·12 (3)	—	2·30 (11)	—	1·95 (7)
17½	11	2·35 (4)	—	2·62 (12)	—	1·75 (8)
Average values		2·26	2·49	2·39	1·95	1·91

TABLE 8. FATIGUE STRENGTH REDUCTION FACTORS, K_f: FILLET SPECIMENS

Superimposed static stress, tons per sq. in.		Values of K_f (uncorrected)				
Bending, f_s	Torsion, q_s	$\theta = 0$	$\theta = 29{\cdot}7$	$\theta = 53{\cdot}1$	$\theta = 75{\cdot}9$	$\theta = 90$
0	0	1·68 (1)	1·77 (13)	1·53 (9)	1·37 (14)	1·37 (5)
17½	0	1·59 (2)	—	1·54 (10)	—	1·28 (6)
0	11	1·52 (3)	—	1·63 (11)	—	1·28 (7)
17½	11	1·75 (4)	—	1·73 (12)	—	1·32 (8)
Average values		1·63	1·77	1·61	1·37	1·30

the form and dimensions of the present fillet specimens:—

K_t (reversed bending stresses) = 2.1.
K_t (reversed torsional stresses) = 1.6.

The experimental values of K_f are listed in Table 8. If the average uncorrected values of K_f for reversed bending and reversed torsional stresses are taken as 1.65 and 1.35 respectively, or $K_f/K_t = 78\frac{1}{2}$ and $84\frac{1}{2}$ percent, the agreement with approximate theory is not quite as close as with the oil-hole. But, as when the oil-hole was present, the previous conclusion again emerges clearly, that the effect of the superimposed static stresses is extremely small, the cyclic stresses being primarily responsible for the reduction in strength.

(3) Six Splines. The case of the splined shaft is of very special interest. When tested under reversed bending stresses, theoretically no stress concentration effect is present: under reversed torsional stresses, a marked reduction in strength should occur; somewhere between $\theta = 0$ deg. and $\theta = 90$ deg., therefore, a marked change, of a discontinuous nature, should occur in the method of failure and the fatigue resistance, not a gradual change as in the case of the oil-hole and fillet. Very definite visual evidence of this was afforded by the method of cracking of the specimens. All specimens tested at $\theta = 0$ deg. and $\theta = 29.7$ deg., entirely irrespective of the nature and amounts of the superimposed static stresses, failed by transverse fatigue cracks which commenced at the crest of a spline and propagated, with slight deviations, on the same transverse plane. But at $\theta = 53.1$, 75.9, and 90 deg., every fatigue crack started at the inner corner of a spline as a shear failure and followed that longitudinal direction for some distance before deteriorating into a transverse fracture.

Cox[30] has made an analytical study of the torsion of long uniform shafts containing discontinuities, including splines. He found that the stress concentration factor at the inner corner of a spline, representing as it does a form of half-groove, is given, approximately by $\left(1 + \frac{1}{2}\sqrt{\frac{d}{r}}\right)$, where d is the depth of the spline and r is the average radius of curvature of its root. In the present case, d is 0.03315 inch and r is 0.0075 inch, giving $K_t = 2.1$: the analysis relates to a long shaft free from restraint; the end restraint present in the combined stress specimens may tend to reduce the value of K_t.

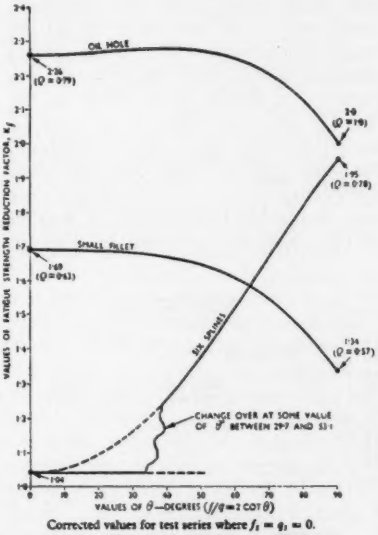
TABLE 9. FATIGUE STRENGTH REDUCTION FACTORS, K_f: SPLINED SHAFTS

Superimposed static stress, tons per sq. in.		Values of K_f (uncorrected)				
Bending, f_t	Torsion, q_t	$\theta = 0$	$\theta = 29{\cdot}7$	$\theta = 53{\cdot}1$	$\theta = 75{\cdot}9$	$\theta = 90$
0	0	1·04 (1)	1·09 (13)	1·47 (9)	1·76 (14)	2·0 (5)
17½	0	1·03 (2)	—	1·56 (10)	—	1·66 (6)
0	11	1·03 (3)	—	1·51 (11)	—	1·86/1·96 (7)
17½	11	1·03 (4)	—	1·50 (12)	—	1·67/1·83 (8)
Average values		1·03	1·09	1·51	1·76	1·83

The uncorrected experimental values of K_f are listed in Table 9. Under reversed bending stresses ($\theta = 0$ deg.), the experimental values of K_f are very near to unity, while the single result at $\theta = 29.7$ deg. also shows almost a complete absence of stress concentration effect; this is in accordance with the observed nature of the fractures. But an unmistakable reduction in strength occurs at $\theta = 53.1$ deg. and a greater one at $\theta = 75.9$ deg.; under reversed torsional stresses ($\theta = 90$ deg.), the fatigue resistance of the material is reduced to, approximately, half its intrinsic strength by the presence of the splines. Comparison with the approximate value of K_t (< 2.1) indicates that the full theoretical effect is approached.

As with the oil-hole and the fillet, the reduction in fatigue strength caused by the discontinuity is primarily due to the applied cyclic stressing; superimposed static stress has little effect.

The actual and comparative effects on the fatigue strength of the material


OIL HOLE
2·26 (Q = 0·79)
3·0 (Q = 1·0)
1·95 (Q = 0·78)
SMALL FILLET
1·69 (Q = 0·63)
SIX SPLINES
1·34 (Q = 0·57)
CHANGE OVER AT SOME VALUE OF θ° BETWEEN 29·9 AND 53·1
1·04
VALUES OF FATIGUE STRENGTH REDUCTION FACTOR, K_f
VALUES OF θ—DEGREES ($f/q = 2 \cot \theta$)

Corrected values for test series where $f_s = q_s = 0$.

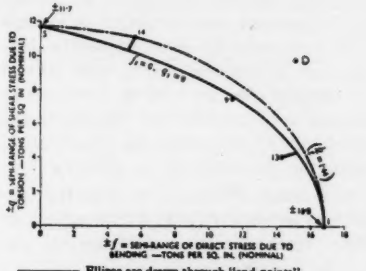
Q = fatigue notch-sensitivity index = $\frac{K_f - 1}{K_t - 1}$.

Discontinuity	Approximate values of theoretical stress concentration factor, K_t	
	Reversed bending	Reversed torsion
Oil hole	2·6	2·0
Fillet	2·1	1·6
Splines	1·0	<2·1

Fig. 17. Fatigue Strength Reduction Factors for Three Forms of Discontinuity (S65A Steel)

caused by the three forms of discontinuity investigated—the oil-hole, the fillet, and the splines—on this S65A material are shown in Fig. 17 where corrected values of K_f are plotted: the corrections have been made by using values calculated on the assumptions that the f/q relation for solid specimens is the ellipse quadrant, the ellipse arc throughout for the oil-hole and the fillet specimens, and a combination of the quadrant and arc —referred to later—for the splined shafts. All three curves relate to series I, Table 6, where $f_s = q_s = 0$.

(c) The Form of the f/q Relation in the Presence of a Discontinuity. (1) Oil Hole. The results of the five tests of series I (Table 6), in which no static stresses were superimposed, are plotted



Ellipse arc drawn through "end-points".
Ellipse quadrant drawn through "end-points".

Fig. 18. f/q Relation for Specimens with Oil Hole (S65A Steel)

in Fig. 18. The relevant ellipse quadrant and arc, drawn through the end-points, are shown. Tests 13 and 14 seem to be irregular but, if this is ascribed to small variations in the material, the ellipse arc certainly provides the best fit for the series and will be taken to represent the f/q relation. There remains for examination the results of series II, III, and IV of Table 6, in each of which three fatigue limit determinations only were made. In each case, the experimental value of the intermediate test point ($\theta = 53.1$ deg.) is compared, in Table 10, with the calculated value of the radial ordinate of the quadrant and arc drawn through the experimental end-points: expressing the results in terms of maximum shear stress is a convenient way of making the comparison. The data fit the arc better than the quadrant: in two cases (tests 10 and 11) the deviation from the arc value does not exceed the accuracy of the deter-

Table 10. Influence of Static Stress (Oil-hole Specimens)

Series	Test No. (see Table 6)	Range of maximum shear stress at fatigue limit					
		Experimental		Ellipse quadrant		Ellipse arc	
		tons per sq. in.	ratio	tons per sq. in.	ratio	tons per sq. in.	ratio
II: $f_s = 17\frac{1}{2}, q_s = 0$	10	±8·6	100	±9·38	109	±8·47	98·5
III: $f_s = 0$, $q_s = 11$	11	±9·0	100	±9·92	110	±9·35	104
IV: $f_s = 17\frac{1}{2}, q_s = 11$	12	±7·7	100	±9·42	122	±8·76	114

mination; the, deviation is more serious by 14 percent, in the third. Although more data must be obtained before definite conclusions are drawn, tests 10 and 11 support the view that when static stresses are superimposed, the ellipse arc satisfactorily represents the f/q relation when the discontinuity is present.

(2) Small Fillet. The five test results of series I (Table 6), in which no static stresses were superimposed, are

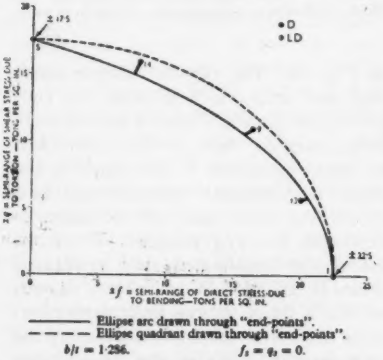
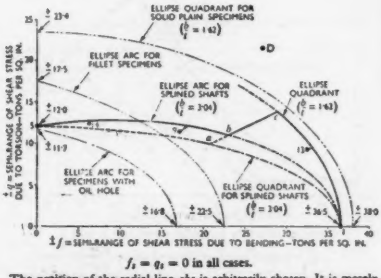


Fig. 19. f/q Relation for Specimens with Small Fillet (S65A Steel)

plotted in Fig. 19. One point (test 13) falls very close to the arc, tests 9 and 14 between the two curves. The ellipse arc provides the better fit and a safer guide to design and may be accepted as a satisfactory relation. The intermediate points of series II, III, and IV, in which static stresses were in operation, are briefly examined, as before, in Table 11. The ellipse arc is definitely the better fit and none of the deviations from the arc values is serious. The ellipse arc thus affords a satisfactory basis for design even when considerable static stresses are imposed.

(3) The Splined Shaft. The five test results of series I (Table 6), in which no static stresses were in operation, are plotted in Fig. 20. As discussed previously, the fractures showed that at values of $\theta = 0$ and 29.7 deg., and also probably up to some higher, and unknown, value, failure did not occur due to the stress concentration effect of the splined contour; hence, from all previous experience of ductile steels tested in the form of solid specimens, tests 1 and 13 should conform to an ellipse quadrant; the same value as found for solid plain specimens, $(b/t) = 1.62$ (Fig. 15*a*) may reasonably be assumed. A portion of this quadrant is shown on Fig. 20



The position of the radial line *abc* is arbitrarily chosen. It is merely known to lie somewhere between the test points 9 and 13.

Fig. 20. Probable f/q Relation for Splined Shafts (S65A Steel)
Other curves assembled for general comparison.

and test point 13 is seen to fall satisfactorily near the curve. In regard to the remaining test points 5, 14, and 9, the value of the end-point t is, of course, given by the value of point 5, but, to compare these data with the corresponding quadrant and arc, some value of b must be taken as the second end-point through which to draw these curves. But no such second end-point is provided by the test data, for, under bending stresses (cyclic and, or alternatively,

Table 11. Influence of Static Stress (Fillet Specimens)

Series	Test No. (see Table 6)	Range of maximum shear stress at fatigue limit					
		Experimental		Ellipse quadrant		Ellipse arc	
		tons per sq. in.	ratio	tons per sq. in.	ratio	tons per sq. in.	ratio
II: $f_1 = 17\frac{1}{2}, q = 0$	10	±13·5	100	±13·6	101	±12·7	94
III: $f_1 = 0,\ q = 11$	11	±12·7	100	±14·7	116	±13·6	107
IV: $f = 17\frac{1}{2}, q = 11$	12	±11·7	100	±12·6	108	±11·4	97

static) fracture is not influenced by the discontinuities of the section of the splined shaft; in fact, as θ changes from 0 to 90 deg., two quite different kinds of fatigue damage are, in turn, encountered. In this difficulty, the only other end-point available—test 1—has been used and a quadrant and an arc drawn in between this and test point 5. The data of tests 9 and 14 fall between the two curves, with either of which they could be associated with small error. All previous tests on specimens containing a discontinuity have conformed to an f/q relation expressed by an arc, but whether the arc shown on Fig. 20 is the correct one certainly cannot be decided from the present tests. If, for the time being and without prejudice to future more definite findings, it is accepted that a portion of that arc represents those tests in which failure occurs at the inner corners of the splines, the fatigue behavior of the splined shafts over the whole range is probably represented by that portion of the arc* extending from test point 5 to "b," followed by an abrupt change to the portion of the "solid" ellipse quadrant extending from "c" to the test point 1. The position of the radial line "abc," Fig. 20, has been chosen arbitrarily: all that is established is that it is situated somewhere between tests 9 and 13.

A much closer experimental and analytical study of this interesting practical case would be well worthwhile.

In these circumstances, an examination of the three isolated results in which static mean stresses are in operation can have little real value but, to afford a comparison similar to that already given for the oil-hole and fillet, the calculated data of Table 12 are recorded. Test 10 practically falls on the quadrant, tests 11 and 12 fall intermediate between the quadrant and arc and are about 5 percent only from either curve. It might, perhaps, be considered surprising if this

TABLE 12. INFLUENCE OF STATIC STRESS (SPLINED SHAFTS)

Series	Test No. (see Table 6)	Range of maximum shear stress at fatigue limit					
		Experimental		Ellipse quadrant		Ellipse arc	
		tons per sq. in.	ratio	tons per sq. in.	ratio	tons per sq. in.	ratio
II: $f_1 = 17\frac{1}{2}, q = 0$	10	±13·3	100	±13·5	102	±14·9	112
III: $f_1 = 0, q_1 = 11$	11	±13·7	100	±12·9	94	±14·4	105
IV: $f_1 = 17\frac{1}{2}, q_1 = 11$	12	±13·5	100	±12·7	94	±14·3	106

particular form of discontinuity did not fall into line with the groove, oil-hole, and fillet in the respect that the ellipse arc affords a satisfactory design basis, but it will be apparent that the splined shaft is a very special case and, moreover, it is not clear which ellipse arc it is justifiable to adopt.

In closing, it may be mentioned that the fatigue strength of an assembly of a splined shaft with its mating member, in which the drive is transmitted from one component to the other, may be less than that of the splined shaft alone: the stress distribution in the two cases is quite different. A few experiments of this type have been made[31] and give an indication of the different behaviors that may occur; more work in this field is required.

The f/q curves relating to the solid plain, the fillet, and the oil-hole specimens are also indicated on Fig. 20; the assembly shows at a glance the relative reduction in fatigue strength caused by the three forms of discontinuity.

I hope that I have not imposed unwarrantably on your time and attention by discussing, in a Presidential Address, such a limited portion of the field of engineering science. If so, I trust that I shall be partly excused on the grounds that the investigation described does break some new ground and that some general conclusions appear to have been established which, used in conjunction

* Or by the portion of the quadrant extending from test-point 5 to "a", or by some curve intermediate between the quadrant and arc.

with the vast amount of fatigue data available in the literature, should assist the designer of engineering components subjected to the four stress variables which have been examined: information has also been provided on the modifying influence of some common and unavoidable changes of section and contour encountered in such components. This investigation has involved considerable cost, testing facilities, and time, yet a commencement only has been made on the general problem. Further investigations of these four variables on other materials, of other stress combinations, especially, of size effect, should be made. But it appears doubtful whether any single laboratory could complete a program of this magnitude within a reasonable period, and, to achieve satisfactory progress and avoid the possibility of wasteful duplication while obtaining useful cross-checks on selected basic data, I would offer a plea for planned cooperation between several laboratories interested in this subject.

ACKNOWLEDGMENTS

In carrying out this investigation from its conception, I have received much valuable advice and assistance of many kinds and from many sources, of which I am making full acknowledgment elsewhere.[6] On this occasion, I wish to mention specially my thanks and appreciation to the Department of Scientific and Industrial Research, to the Air Ministry, and to the authorities of the National Physical Laboratory for the constant support and generous grants received and for the facilities made available in carrying out this lengthy investigation and for permission to publish the results; also to my joint investigators, Mr. H. V. Pollard and Mr. W. J. Clenshaw, former colleagues at the N.P.L. and fellow-members of the Institution; also to the Controller of His Majesty's Stationery Office.

Figs. 6 to 20, inclusive, of this address are Crown copyright reserved and are reproduced with the permission of the Controller of His Majesty's Stationery Office.

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THE ENGINEER AS A MANAGER

ACKNOWLEDGMENT

Mechanical Engineering, April 1950, contained this article by Mr. J. M. JURAN, Chairman, Department of Administrative Engineering, New York University. The article discusses the relationship of management to engineering, what the engineer must do to become a manager, and the training of engineers in management. It was presented at the Management Conference under the auspices of the Metropolitan Section and the Management Division of The American Society of Mechanical Engineers, New York, N. Y., Oct. 1949.

More than half of the young engineers of today aspire to end up as executives or managers. Less than half of the young engineers aspire to end up as scientists, consultants, teachers, or other nonmanagers.

The young engineers who would be managers aspire not merely to become managers of engineering departments. They hope to become general managers, officers, and directors of companies, in capacities which oversee not only the activities of engineers, but of others as well.

In these aspirations, the young engineers are not alone. The young accountants, lawyers, salesmen, mechanics, and many others likewise hope to become managers.

This keen and extensive competition to occupy managerial posts is a sign of increasing alertness to the vital role of the manager in the industrial society. In past centuries, leadership in the economy has been held variously by the soldier, the merchant, the priest, the philosopher, the financier. Today, and for the forseeable future, the manager dominates the economy.

It is the purpose of this paper to analyze the qualifications of the engineer for managerial posts. To make this analysis it is necessary to consider the following:

(a) What is the relationship of management to engineering?

(b) Does engineering training qualify one to become a manager?

(c) What must the engineer do to become a manager?

RELATIONSHIP OF MANAGEMENT TO ENGINEERING

The skills needed to perform the tasks of modern industrial society are exceedingly varied. They include the numerous mechanical skills exercised in the factory, office, warehouse, on the farm, on the railroad. They include the professions. They include, of course, the skills of the scientist and of the engineer.

But the skill of the manager[1] is different from any of these. Essentially, the role of the manager is to get things

[1] The term "manager" is used here in its generic sense as one who must act through the efforts of other people.

done through other people. To carry out this role, the manager requires skills which are as yet not well understood, and certainly not well executed. The knowledge of "how to manage" is only now being developed. The art of management today stands perhaps in the situation that understanding of electricity stood in the days of Volta, Galvani, and Ampere.

The distinguishing feature of engineering is the use of "the forces of nature" for the benefit of man. The distinguishing feature of management is the use of "the forces in people," also for the benefit of man.

The forces of nature are discovered and measured by the "natural" scientists —the mathematician, the physicist, the chemist. The engineer utilizes for the benefit of man, the knowledge discovered by these and other natural scientists.

The manager, in contrast, uses the forces of human beings, also for the benefit of man. The forces of human beings are discovered and measured by the "living" scientists, the biologist, the psychologist, the sociologist, and others.

To utilize the forces of nature, the engineer has evolved various bodies of engineering knowledge: thermodynamics, machine design, electrochemistry, and many others. These bodies of knowledge are engineering tools. Success in engineering depends in good measure on the extent to which the engineer can grasp and use these engineering tools.

Similarly, to utilize the forces of human beings, the manager has evolved various bodies of managerial knowledge: organization, controls, incentives, and many others. These bodies of knowledge are management tools. Success in management depends in good measure on the extent to which the manager can grasp and use these management tools.

However, there is a further consideration, one of the utmost importance. We cannot utilize the forces of nature, except through the medium of human beings. A machine designed in accordance with engineering principles must still be constructed, operated, and maintained by human beings. (The engineer who designs it is himself a human being.) This is the thinking behind the 1913 ASME definition of management:

"Management is the art and science of preparing, organizing, and directing human effort applied to control the forces and to utilize the materials of nature for the benefit of man." [2]

The foregoing explains, in the author's opinion, why the manager so universally "outranks" the professional man in popular contemplation. The manager outranks the engineer. The dean outranks the professor. The director of research outranks the scientist. The general outranks the ballistics expert. The bishop outranks the priest. And so on and on. The manager outranks the professional man because it is so universally recognized that utilization of the forces of nature can be done only through human effort.

CASE HISTORY OF AN ENGINEER WHO BECAME A MANAGER

In making these distinctions between engineering and management, I hasten to add that relatively few jobs involve "pure" engineering or "pure" management. We can trace the career of a young engineer to see how imperceptibly he glides from engineering into management.

In 1922 Bill was a young mechanical engineer fresh out of engineering school. Bill was ambitious, but not to a point of fanaticism. He decided he was going to join a good company as a second lieutenant, climb the ladder, rung by rung, and retire on a colonel's pension some 40 years later.

[2] Transactions ASME, 1913, p. 1272.

So Bill joined the X company, one of the industrial giants. Bill was on a training course for almost a year, going around from department to department, learning about the business. Then he was given a job as assistant engineer in one of the product-design departments. He was on the drafting board for a few months, then prepared bills of material for a few months, then had a job of doing odds and ends in the drafting room. In 1925 he was designated "engineer," and became one of a flying squadron of engineers who were assigned, first to this project engineer, then to that project engineer, figuring stresses, designing components, choosing materials, and performing other engineering chores. Bill did these things until his promotion to project engineer in 1928.

Until that promotion, Bill was almost all the time using the forces of nature for the benefit of man. Most of the time Bill was using Young's modulus. Ohm's law, other laws, some calculus, lots of arithmetic. The visible evidences of Bill's work were in sketches, calculations, memoranda. Bill's engineering training was very helpful in doing all of these things.

But Bill was also dealing with people. He had various bosses in the form of project engineers. He worked with associates under the same boss. He worked with shop supervisors, with draftsmen, with production expediters, with accountants, and with still others. Bill's engineering training was devoid of any suggestions on how to work with a boss or with these other people.

During the latter part of this time Bill had an assistant—a young fellow just out of college, who was assigned to Bill for training. Bill was able to get this trainee to take data, make calculations, prepare drawings. Here Bill was achieving things not just through his personal efforts—he was utilizing the force of a human being. He was, on a small scale, managing.

As project engineer, Bill's duties changed. He now spent much less time in calculations or in sketching—the engineers who "reported" to him did those things. He assigned work to them, reviewing the work as it progressed. He now had to make broader decisions, but still largely of an engineering character, still of a character which utilized the forces of nature.

But as a supervisor, Bill now had also a set of duties concerned with human beings. Each person was different. Ed, the old-timer, was willing to work but was in bad health. Frank was about Bill's age, and was burned up because he thought that he rather than Bill should have been made project engineer. Steve was another tough case—he was openly fed up. The company hadn't treated him square. And so on and on. How to get these men to work wasn't in the engineering books. How to get these men to work together wasn't in the books either.

The relative ease of commanding "things" was in sharp contrast to the difficulty of commanding people. "Things" could be pushed around by force. People couldn't be pushed around like that. People had to be persuaded.

We skip over the headaches of the depression of the early thirties and the recovery. We credit Bill with much foresight for his decision to go back to school and take night courses in management. In these courses he discovered that these new management problems of his were very old, that they had names, that they had been analyzed, and that there existed ways of solving them.

It is 1937 and Bill is chief project engineer for the whole division. Bill still makes decisions of an engineering character, but now strongly flavored with other considerations. Most of his time is now not on engineering problems at all. He spends much time in meetings with other department heads. Their discussions and decisions involve many

varied problems—accounting, patent law, collective bargaining, sales forecasts, advertising, government regulations. Most of these discussions seem quite remote from engineering. Now when Bill wants to get something done, he assigns it to a project engineer, or he asks some other department head to get it done for him.

In 1945 Bill became engineering manager for his division. Now he was top dog among the engineers, but he did very little engineering. His calculus had long since rusted into disuse. He remembered few formulas. His time was spent in looking over budgets, attending conferences, approving memoranda and reports, rating subordinates, performing numerous personnel duties.

Bill isn't through progressing. The general manager of the division is due to retire, and Bill will become general manager of the division. Then he will become responsible not only for engineering, but for manufacture, and sales as well. All pretense at utilizing the forces of nature will be abandoned. Especially when it comes to sales and manufacture, Bill will have to rely absolutely on the experts—on the men down the line who are supervisors, accountants, market analysts.

Before Bill retires he will have become a vice-president of the company, and a director. In these posts he will be in the cabinet. He will contribute to discussion and decision affecting all phases of company operation—not merely those phases for which he is officially responsible. The transition from engineer to manager will be complete.

In following Bill through his success story, we see that while the clevage between management and engineering is abrupt enough in theory, the transition in practice is most gradual.

DOES STUDY OF ENGINEERING QUALIFY ONE TO BECOME A MANAGER?

Our definition is that the manager is one who utilizes the forces in human beings for the benefit of man. Engineering provides no training on what are the forces in human beings, let alone how to use these forces. We must conclude that study of engineering does not qualify one to become a manager.

But neither does study of law, medicine, accounting, or any other one of the professions specially qualify one to become a manager. Does it follow that the young engineers, lawyers, or accountants who aspire to become managers all went to the wrong schools? What should they have studied?

The answer is that there is today no school which purports to turn out men capable of stepping into managerial posts. If there were such a school, it is most unlikely that the graduates would secure placement as managers. For while industry has no scientific method for identifying good managers, it does know that good managers must possess certain special qualities such as balance, maturity, and leadership, and that to acquire and season these qualities requires much time and experience as well as training.

Today there is not even an agreed-upon program of study for potential managers. For the most part the potential managers must learn from their elders, perpetuating their good or poor practices, just as did the lawyers and doctors of a century ago.

Does it mean that engineering is to continue doing business at the old stand, leaving engineers to gravitate into managerial posts as best they can? If the engineering profession concluded this, it would be a disservice to the aspirations of a majority of its members. The engineering profession should, rather, adopt a course of action which would aid

engineers to qualify themselves more readily for managerial posts.

I do not advocate this on the familiar contention that the engineer, having built the machine, is thereby best qualified to oversee the Machine Age. In fact, I disagree with that contention. My argument is rather that the engineer, aside from his knowledge of the machines, is trained in a methodology sorely needed in management.

The engineer builds structures and machines through the engineering method of scientific utilization of natural laws previously discovered by the natural scientists. Built in this way, the structures stand and the machines run. Failure is so rare as to be newsworthy.

The need in management is to build organizations of human beings to run so well so often that failure likewise becomes so rare as to be newsworthy. An important prerequisite for meeting this need is scientific utilization of the forces of human beings as these forces are discovered by the "living" scientists. The engineering method is sorely needed in management, though applied to knowledge of the living sciences rather than the natural sciences.

Returning to the question, "Does engineering training qualify one to become a manager?" we can now say that engineering training can be most helpful because it supplies an important methodology, that of the engineering method of decision based on knowledge derived from reproducible experiment. However, this methodology is not enough. True, the engineer is all dressed up with a methodology which is valuable to the solution of management problems. But the engineer as such has no place to go except in engineering, because his knowledge of the living sciences is not adequate for the managerial task.

I believe this one explanation why many engineers, despite their admirable logic, remain technicians while the arts-school graduates, the lawyers, the salesmen seem to go on to become chief executives. (Incidentally, some engineers also end up in the high circles.) These rivals, though not so strictly schooled in the discipline of the engineering method, are generally better equipped with knowledge of the living sciences. Thereby many of them have a better grasp of the thing we call human nature.

WHAT MUST THE ENGINEER DO TO BECOME A MANAGER?

Again we return to the definition of management—the utilization of the forces of people for the benefit of man. Implied in this definition is the knowledge of:

(a) What are the forces of people?
(b) How are these forces utilized?
(c) What is for the benefit of man?

These questions are so fundamental in character that it may well be contended: "Surely the managers of today do not know these things and yet they get along. Is this knowledge then essential?"

It is true that only rarely does a manager of today know all these things. But it is stretching a good deal to say that the managers "get along." The mortality rate of business enterprises is shockingly high. We can build bridges, generators, aircraft, with the highest confidence that they will perform per specification. But we have no such confidence when we launch a new business enterprise, or when we set up a new department in an existing enterprise.

Today, the engineer who seeks to prepare himself for a managerial post is on his own volition turning to the graduate schools of Administrative Engineering and of Business Administration. There he studies psychology, sociology, and others of the living sciences to learn more of man as an individual and

of men as a group. He studies various tools of management—organization, controls, incentives—to learn how to utilize these human forces; and he studies economics, law and other disciplines to learn what is for the benefit of man.

All this is progress. Yet it seems to me that as the tempo of the industrial society steps up, the caliber of the manager must rise remarkably. In particular, the manager must see in proper perspective the relation of all the living sciences to each other. Only through such perspective can he perform the trusteeship job which is his duty if he is to emerge as a member of a profession.

SHOULD THE DEFINITION OF ENGINEERING BE OPENED UP TO INCLUDE MANAGEMENT?

It has been advocated by some that we should change the definition of engineering to read: "Engineering is the utilization of the forces of nature and of human beings for the benefit of man." This would not be the first time that engineering has increased its scope.

But can management, which rests on the living sciences, become a branch of engineering, which rests on the natural sciences? I doubt it. To stretch engineering to include the living sciences, it seems to me, goes beyond the elastic limit.

In my opinion, it is most desirable that the engineering profession be as well represented in top management as are the other professions and the arts. The industrial society is bound to run askew if policymaking and execution is dominated by the banker, or the lawyer, or the labor leader, or for that matter, by the engineer.

The risk today is not that management will become topheavy with engineers. On the contrary, the risk is that engineering training, with its great emphasis on the natural sciences, will develop increasingly a "trained incapacity" on the part of engineers to advance into managerial posts. Years of commanding kilowatts and decibels, and of securing invariable compliance, can be sorry training for a managerial post where command means so little and leadership means so much.

TRAINING ENGINEERS IN MANAGEMENT

The engineering colleges took a long stride toward broader development of engineers by adopting the so-called humanities courses. Until the adoption of the humanities courses, the undergraduate engineers were so closely confined to the study of things, that there was little time for study of people.

I doubt that much more can be done at the undergraduate level to prepare engineers to be managers. True, the undergraduate can learn something about industrial-engineering tools. But he has neither the experience nor the maturity to grasp firmly the nature of the managerial problem of doing things through other people. For the forseeable future, the opportunity for training engineers in management lies in graduate study and in industry itself.

Opportunity for such graduate study in the colleges is expanding rapidly, both in the variety of courses and in the number of colleges offering such courses. The interest on the part of graduate students is likewise expanding rapidly.

But in the immediate future the biggest role in training for management is to be played by industry itself. A serious

limitation on college training in management is the lack of an adequate laboratory. In study of management, the laboratory is industry itself. To utilize this laboratory suggests that industry must participate heavily in the job of training. In time we will work out a decidedly better collaboration between industry and the colleges. Meanwhile, the fact that industry has so nearly a monopoly on the management laboratory puts a special duty on industry to identify prospective managers, and to help teach them how to manage.

For industry to teach men how to manage still requires close collaboration between industry and the colleges. Industry abounds in case histories, but discovery of principles from these cases requires a forum which can collect the experience of many companies.

In our graduate courses and in our management round tables at New York University we bring together men from a variety of industries to study various aspects of management. The college provides the forum but industry provides the cases.

However, we are far from having discovered all the possible devices for teaching management to men in industry. For example, there are current experiments in England wherein a man works in industry four days a week and spends the fifth day in graduate study at college. In some of the American companies, men of managerial promise are given special training through rotation, through participation in committee work, through schemes of "multiple management," through assignment to staff positions, and through still other means. The experience we are gathering will soon permit us to appraise the effectiveness of these various devices for management training.

CONCLUSION

When we look back on the history of professions, we see that the rise of any true profession has heaped its bounty on humanity. Some of the more obvious instances have been truly dramatic. In our century the rise of the medical profession has greatly increased our span of life. The rise of the engineering profession has taken heavy burdens from the backs of men, and placed them on the backs of machines. The collective efforts of all professions have wiped out starvation, increased our standard of living, and have added much leisure time in which to enjoy that standard.

Yet there remain grave problems in the industrial society. Mysteriously ahead of us loom the depressions, the industrial struggles, and other ominous shapes. Our forbears must have regarded famine or pestilence with equal mystery and apprehension.

The challenge to the manager is to solve these problems of the industrial society. If he fail, he must make way for new leadership, as did the financier before him. If the manager succeed, he will have assured for himself a place of honor among the professions.

The engineer is possessed of an intimate knowledge of the machines of the industrial society. He is also possessed of a proved methodology for solving any problem in the sciences. This knowledge and this methodology can be of great aid to the select managerial circle. It behooves the engineer to supplement his engineering knowledge by such study as will enable him to compete successfully for a reasonable proportion of the managerial posts. Thereby, he will not merely satisfy his own aspirations. he will perform a vital and honorable public service.

THE OUTLOOK FOR CERAMICS IN GAS TURBINES

ACKNOWLEDGMENT

This article is by Messrs. W. H. DUCKWORTH and I. E. CAMPBELL, Assistant Supervisors, Battelle Memorial Institution. It appeared in *Mechanical Engineering* for February, 1950. It discusses the use of ceramic bodies as rotor blades, stator blades, combustion chambers, etc., and the employment of ceramic coatings.

INTRODUCTION

New heat-resistant materials are being developed continuously through ceramic research. Part of the research is directed specifically toward gas-turbine applications, and much of the remainder is in some way related to such uses.

Not everyone has the same conception of the ceramic field. For purposes of discussion "ceramics" embrace all that is not clearly organic or metallurgical. To be more specific, it is probably advisable to cite some of the materials which fit under this definition, and which are being considered in the present research. There are of course the old, conventional, silicate-based materials, or those composed of, or bonded internally with glass. Less conventional materials, which do not contain glass, also are included. Many of these glass-free materials have, as primary constituents, one or more of the refractory oxides, carbides, borides, nitrides, and the like. Others are composed essentially of carbon or graphite. In addition, the field includes materials composed of various combinations of metal and inorganic compounds. The metal may be present in these hybrid materials as either a continuous phase or as isolated inclusions.

There are innumerable raw materials and combinations of raw materials from which ceramic articles may be fabricated. Many raw materials and combinations of raw materials remain wholly uninvestigated. They are not being overlooked, but have been neglected because not enough time or effort has yet been spent in research.

HISTORY OF PERTINENT CERAMIC RESEARCH

The pertinent research was initiated during the war. It is part of the effort directed broadly toward the discovery, development, or refinement of materials to meet temperature requirements peculiar to propulsion of high-speed high-performance aircraft and guided missiles. Prior to this interest in ceramic materials, applications for these materials did not involve designs where shape and dimensions were so critical, or conditions of stress and temperature nearly so severe. To illustrate the marked differences in past and present demands for ceramics, the contrast between service conditions for spark-plug insulators, wall tile, refractory brick, or metallurgical crucibles, and those for rotor blades might be considered as representative.

As might be expected from the considerable differences between past and present demands, ceramic materials available when this research was started generally were not adaptable to the new uses. Unfortunately, the adjustments needed to improve their likelihood of success were rarely simple ones. Some mention was made previously of the fact that these conventional ceramic materials are characterized by the presence of a glass phase. It so happens that glass has notably poor resistance to thermal shock and tends to soften and deform under stress at relatively low temperatures. Even if a ceramic material is almost wholly crystalline, containing glass as only a contaminant, the glass phase may interfere with the continuity of the crystal phase and serve as a "lubricant" under stress at high temperature. The glass phase can hardly be tolerated in structural parts for the hot zones of gas turbines.

The elimination of glass from the final product necessitates excluding silicates from the raw materials. Clay, among other raw materials, cannot be used. Ceramists utilize the plastic properties of clay in forming articles. After forming, the clay bonds the nonplastic particles so that the shape has sufficient strength for necessary handling until it is fired. Further, during firing, the clay enters into thermochemical reactions which develop the fired or glass bond. The properties of the final article depend largely upon the nature and extent of these reactions. Clay has a big role in each major step of the fabrication process.

GLASS-FREE REFRACTORY MATERIALS

The development of a ceramic material for a specific use might be divided conveniently into three main phases of research, (1) raw-material selection, (2) forming, and (3) heat-treating.

The initial selection of raw materials is based primarily on melting point, stability, and availability. There are many raw materials and combinations of raw materials which remain unexplored. In glass-free ceramics, where silicates must be excluded as raw materials, the supply still greatly exceeds present research demands. This favorable situation probably will persist for a long time. Investigators are continually uncovering additional refractory compounds of interest, and it is a quite lengthly procedure to develop an optimum body of a selected composition.

Investigations pertinent to gas turbines have been rather thorough on most of the refractory oxides singly, and on many combinations of oxides. There is room for much more research, however, on combinations of oxides. With a few exceptions, the development of bodies from other likely raw materials has been preliminary in nature, or has not been undertaken. The

exceptions mainly involve some combinations of oxides and metals, oxides and carbides, and developments stemming from the background on carbide cutting tools. In the latter category, the development and evaluation of mixtures of titanium carbide and cobalt for rotor blades is well advanced. In general, the surface has only been scratched in the detailed development of bodies of refractory carbides, nitrides, borides, silicides, and the like, singly, in combination with each other, or combined with metals.

It is not surprising that the oxides have received more attention than the other raw materials. They are more available or more easily prepared, and of course more stable in the oxidizing atmospheres usually encountered in service.

After selection of the raw materials, the next major problem is to form them into some desired shape. Considerable success has been had in research along this line. The raw materials for forming normally consist of fine nonplastic powders. The use of various organic compounds with these powders, as bonding and plasticizing agents, has apparently progressed to a point where they are commensurate with clay for ceramic forming. It is no longer much of a problem to use any of the conventional ceramic-forming techniques of dust pressing, plastic extrusion, or slip casting in the fabrication of shapes of such inorganic and organic mixtures. Further, it is often possible to produce these shapes so as to have sufficient strength for easy machining while in the green unfired state.

Finally, after appropriate curing or drying, the formed shape is subjected to a heat-treatment or sinter. The changes which occur in the internal structure during sintering determine the properties of the product. The mechanism of sintering is not fully understood at present, and the action is influenced greatly by several factors. For these reasons, development is hindered most in this phase of research.

Normally, during the early stages of heat-treatment, any organic materials present are oxidized and removed, or are volatilized with a carbon residue remaining. On further heating, the crystalline particles usually coalesce, and chemical reaction, solution, recrystallization, or grain growth may occur. The nature and extent of the phenomena which occur in the structure do not depend solely on the composition, and the time and temperatures of sintering. They are also affected, and often to a great extent, by such factors as impurities and minor additives as well as the major constituents, the grain size and character of the raw material, the forming method used and degree of compaction during forming, and the atmosphere during sintering. The influence of these many factors makes it necessary to carry out an extensive program in order to arrive at the best internal structure for a selected composition and use. However, more is learned every day about the sintering process. Knowledge is being acquired on how the bond between particles is developed, and on the manner in which the bonding action is influenced by various factors. With this increasing knowledge, the problems in developing ceramic materials for gas turbines are becoming simpler.

EVALUATION OF CERAMIC BODIES AS ROTOR BLADES

In general, some of the appraisals of ceramic materials are empirical, based on service testing, while others are theoretical, based on property determinations. The two approaches of course complement each other, and the sole use of either would be impracticable.

The applicability of ceramic bodies as turbine-blade materials has been de-

scribed previously by Bobrowsky.[1] In his discussion of the subject he covered evaluations based on both property determinations and service testing, pointing out that several ceramic bodies had higher ratios of tensile strength to density than the better commercial alloys at temperatures in the range of 1500 to 2200 F. Creep rates at 1800 and 1900 F apparently were adequate for oxide bodies, the only ones upon which creep data were available. He warned that oxidation problems and difficulties with wheel cooling may be encountered in the use of ceramic bodies with high thermal conductivities, while lack of sufficient thermal-shock resistance may limit seriously the use of low-conductivity bodies. Normally, the nonoxide raw materials and metal incorporations contribute to high conductivity, while low-conductivity bodies have high oxide contents. In service tests, blades of both an oxide body and of a metal-bonded titanium-carbide body were operated at temperatures above those presently used with alloy blades, but at lower rotor speeds, and under less severe thermal-shock conditions. These service tests revealed certain problems in the use of the ceramic bodies. There were stress concentrations between wheel and blade when the ceramic blades were substituted directly in present metal designs. Severe thermal shock, resulting from failure of the air supply, caused fracture of blades in one instance. Mechanical-shock problems also were encountered.

It is interesting to note that each shortcoming of the ceramic blades in these service tests is traceable to lack of ductility. If a ductile body exists, we are not aware of it. Also, the possibilities of developing such a body seem remote. This same situation apparently exists with new high-temperature alloys. Sweeny,[2] writing about turbine alloys, stated: "The metallurgist can produce stronger materials at a given temperature, or equally strong at a higher temperature, if the mechanical engineer can learn to fabricate and assemble materials with lower ductility. Close co-operation will be required in the years ahead between the metallurgist and the engineer in the development of new and better alloys that can be adapted to gas-turbine designs." These quoted remarks appear equally applicable to ceramists, and, in the interests of promoting the closer co-operation between ceramists and gas-turbine engineers in the future, it might be well to mention some factors which we feel are significant.

ROLE OF GAS-TURBINE ENGINEERS

As an outlook, several ceramic bodies, presently available, might have adequate properties to permit increases in the operating temperature or stress, or both, on present rotor blades. Better bodies can be expected in the future. All of these bodies, however, are nonductile; hence mechanical shock and stress concentrations are expected to present critical problems in their use. Accompanying the brittleness, some of the bodies also have serious thermal-stress limitations. Maximum performance from ceramic bodies as rotor blades therefore depends on design and operational adjustments to counteract this brittleness. These adjustments, for the most part, involve problems in gas-turbine engineering beyond the ken of ceramists.

Further, evaluations based solely on property determinations shed little light on the influence of lack of ductility on service performance. For example,

[1] "The Applicability of Ceramics and Ceramals as Turbine-Blade Materials for the Newer Aircraft Power Plants," by A. R. Bobrowsky, Trans. ASME, vol. 71, August, 1949, pp. 621-629.

[2] "Wanted: Better Criteria for Turbine Alloys," by W. D. Sweeny, *Metal progress*, vol. 55, 1949, p. 315.

thermal-shock requirements for a given use cannot, at present, be related to the thermal-shock properties of a material. The requirements depend on design as well as time-temperature relationships, and the properties appear to depend on the magnitude of several other physical properties at the time of failure. Some sort of simulated service test therefore is required for adequate appraisals. The need for strictly gas-turbine engineering is apparent here. According to Sweeny,[2] again we find this same problem in metallurgy. Paraphrasing his words, we can say, ". . . both the ceramist and the engineer have a fundamental challenge of a different sort—that is, to develop a laboratory test for evaluating materials that correlate closely with service experience. Until such a measuring stick becomes available, the full development of better bodies will be impaired."

Becoming somewhat less practical, there is another consideration where close co-operation between ceramists and gas-turbine engineers might be beneficial. Ceramic bodies are inherently stronger in compression than in tension, as well as inherently brittle. The compressive strengths of some of the oxide bodies are of the order of 10 times their tensile strengths. This of course indicates that the best design would have ceramic parts either prestressed in compression, or subjected to compressive loads. Apparently, little has been done along this line. If suitable designs could be evolved, prestressing offers possibilities of reducing thermal and mechanical-shock problems, as failures from such shocks probably are basically tensile in nature. If, on the other hand, the blades are subjected to compressive loads, porous bodies might be used. The bodies would have lower strengths, but better shock resistance than similar dense bodies.

EVALUATION OF CERAMIC BODIES FOR STATOR BLADES AND COMBUSTION CHAMBERS

With lower mechanical requirements, stator blades offer a somewhat more likely future for ceramic bodies than rotor blades. Ceramic stator blades, made from bodies too weak for rotor blades, but with good thermal-shock resistance, might be satisfactory. Even without higher operating temperatures, it is possible that such blades would reduce weight and cost if used to replace metal blades in present designs.

As in the case of rotor blades, physical properties cannot, at present, be related to service requirements in a manner suitable for precise theoretical appraisals. However, as approximations, a few possibilities might be discussed. Bodies which are strong enough to withstand the thermal shock by virtue of being porous, may exist. There is sufficient background on oxides, such as magnesia, alumina, or zirconia, and on zircon for their quick evaluation as primary constituents of such bodies. In view of the considerable progress in ceramic research on protecting graphite bodies against oxidation, these also are possibilities. Certainly they would resist the thermal shock and, perhaps, the corrosion and mechanical stress. Although limited in temperature resistance, fused silica and cordierite bodies should warrant detailed evaluation because of their low thermal expansion resulting in good resistance to thermal shock. They might be satisfactory for present operating temperatures, or slightly above. High-conductivity bodies, such as those of silicon carbide or titanium carbide, also have good shock resistance and may work.

This list of potential stator-blade materials undoubtedly could be extended. It is believed sufficient, however, to show that there are several ceramic approaches to the problem. The economic

advantages are also emphasized, as all of the bodies mentioned are prepared from readily available raw materials and have much lower unit weights than the heat-resistant alloys.

Active, but rather limited, consideration is being given ceramic stator blades in simulated-service tests. Unlike the rotor-blade tests, the thermal-shock conditions in these tests are as rigorous as those in present turbines. The early results do not permit definite recommendations, but are considered sufficiently encouraging to predict that improved stator blades can be made of present-day bodies with little or no refining.

Ductility undoubtedly is necessary in the combustion-chamber liners of present designs, thus eliminating direct substitution of ceramic bodies for the sheet metal now used. There are indications that metal-supported ceramic liners will withstand the conditions of service. Also, ceramic bodies might be used to insulate combustion chambers if the need becomes sufficiently acute to justify design changes. Considerable success has been had in so insulating much hotter rocket chambers.

CERAMIC COATINGS

In addition to the work on ceramic bodies, a large portion of the ceramic research pertinent to gas turbines is directed toward developing and evaluating refractory ceramic coatings for metals. Such coatings seem particularly attractive for immediate use. They are not expected to afford outstanding benefits, but they are likely to extend significantly the usefulness of alloys, and few, if any, design changes are anticipated in order to use them. The primary purpose of the coatings, presumably, is to seal the metal surface against corrosive attack, and secondarily to provide thermal insulation. Not much can be expected from them as insulators as they must be thin in order to be tough and adherent.

Most of the work on coatings has been confined to the enamel type. These consist essentially of a thin layer of glass and, as such, are subject to yielding under stress at relatively low temperatures. However, we are not so concerned with this weakness in coatings as in bodies because the base metal, not the coating, bears the stress. Spall-resistant coatings have been developed to "fit" the low-carbon and low-alloy steels, and the significant heat-resistant alloys, with the possible exception of high-molybdenum or tungsten alloys. The refractoriness of some of these coatings is such as to afford protection for long periods at temperatures up to at least 1800 F. In laboratory tests, coated specimens have been unaffected under conditions of time, temperature, and atmosphere which caused complete deterioration of uncoated specimens. In service tests, instances are reported where the enamel-type coatings have increased the life of rotor blades and combustion-chamber liners appreciably. The worth of these coatings, however, has not been established definitely through extensive service testing. It is not known, for example, whether they will permit increases in operating temperatures, or which of several available coatings are better. The reasons for the reputed beneficial effects are not clearly understood. If they were, better coatings for the specific uses probably could be developed. Here, again, close cooperation between ceramists and gas-turbine engineers, along with metallurgists, might pay off.

Another coating development has produced spall and oxidation-resistant coatings of metal-ceramic combinations. Some of these appear to be more heat-resistant than the enamel-type coatings, and more techniques have been developed for their application. Little success

has been had in maturing the enamel coatings by means other than furnace firing, while the metal-bonded coatings can be matured with a torch, or even flame-sprayed. The relative merits of these coatings as compared to enamel coatings have not been established. The characteristics of the two types are rather different, and each may have uses in gas turbines for which it is particularly suited.

As temperature requirements for materials of construction are pushed slowly but continuously upward by design engineers, attention is being given to new techniques for the application of refractory coatings not readily applied by conventional methods. The most generally useful methods developed to date for the formation of refractory coatings involve the so-called vapor-deposition processes in which the coating is formed by chemical reaction at the heated surface of the base material. Commercial application of vapor-deposition processes has, as yet, been limited to materials for moderate-temperature service, e.g., chromized steels, siliconized iron, and so forth, but current research has shown promising results in several applications of interest to gas-turbine engineers.

In vapor-deposition processes, volatile compounds containing the coating elements are passed over the work, which is heated to a temperature at which the components of the plating atmosphere react to form the desired coating. The base, itself, may, and frequently does, enter into the plating reaction, but this is not necessarily the case. Specific examples of processes for the formation of refractory coatings include the deposition of metals such as tantalum, molybdenum, and tungsten by hydrogen reduction or thermal decomposition of their chlorides or bromides, the formation of coatings of silicon carbide, titanium carbide, tantalum carbide, and the like, by the reaction of their chlorides with a hydrocarbon-hydrogen atmosphere, and the formation of silicide coatings by the reaction of silicon tetrachloride-hydrogen atmosphere with a refractory metal base.

A wide variety of chemical reactions is available for formation of protective coatings by vapor-deposition techniques. Coatings of all of the refractory metals, carbides, borides, silicides, oxides, and nitrides can be obtained by one or more of these processes.

Obviously, many of the refractory materials are quite brittle, and differences between their coefficients of expansion and that of a given base may restrict their use. In many cases, however, by proper selection of coating and base materials, increased service life is obtained.

Vapor-deposition processes also can be used to coat ceramic materials such as graphite, alumina, zirconia, and tungsten carbide.

Much of the work on commercial application of vapor-deposition processes is unpublished; hence the application of specific coatings to gas turbines cannot be discussed here. It is important to note, however, that these processes are available, that they are extremely versatile, and that consideration should be given to them where commercial processes are either unavailable or inadequate.

Perhaps the outstanding development on vapor-deposition processes is the development of protective coatings for molybdenum, which have permitted its use in air for over 5000 hr at 1800 F, and as long as 500 hr at 3100 F. In a number of tests for short-time applications in extreme high-temperature flames, molybdenum so protected has given outstanding performance.

Examples of coatings of potential interest are silicon or silicon carbide on graphite for oxidation resistance, boron carbide on graphite and other ceramic materials for wear resistance, and refractory metals, such as tungsten and

tantalum, on other metals for heat and erosion resistance under nonoxidizing conditions.

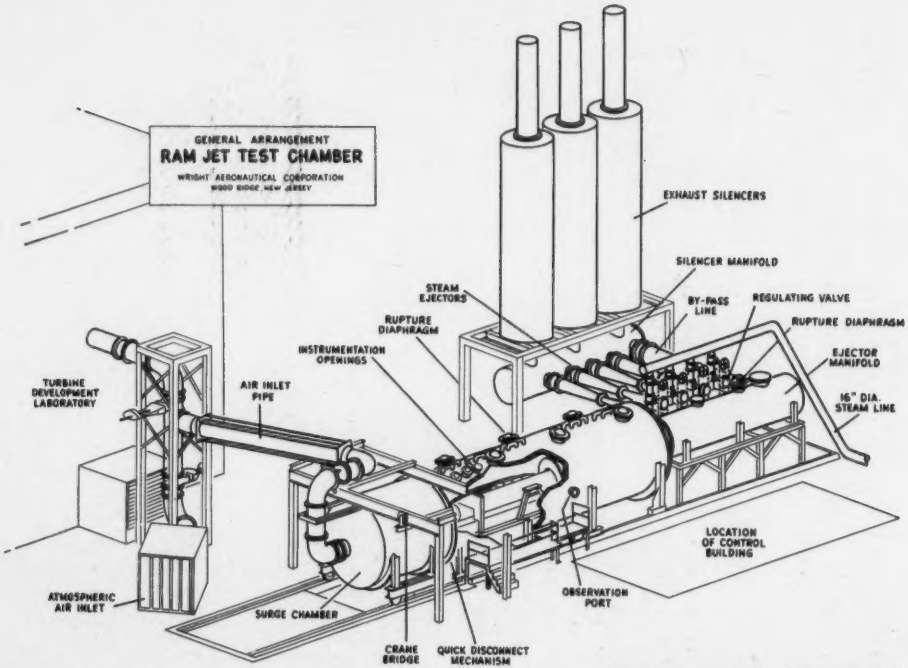
Coatings applied by vapor-deposition methods normally are of theoretical density, and, although subject to localized imperfections if improperly applied, are nonporous as deposited. Hence an outstanding advantage of these processes is that no heat-treating is necessary to obtain a continuous coating. Therefore the coating need not be limited by a maturing temperature which, in turn, is limited by the heat resistance of the base metal. An additional advantage of vapor-deposition coatings is that coatings formed at a high temperature are frequently more stable, at high temperatures, than those formed at room temperature.

CONCLUSIONS

From the standpoint of pure ceramic technology, rapid advances have been made within the past few years in ceramic research relevant to gas turbines. Even so, there are many uninvestigated raw materials and combination of raw materials from which better gas-turbine components might be made. Also, much remains to be learned about the mechanism of sintering in glass-free ceramic materials.

In general, several available ceramic materials might serve advantageously for parts operating in the hot zones of gas turbines. Coatings for metals seem particularly attractive for immediate use, while ceramic bodies might afford more outstanding benefits. Design or operational adjustments often may be necessary to obtain maximum performance from ceramic bodies.

To evaluate and to use all types of ceramic materials, close co-operation is required between ceramists and gas-turbine engineers. With this co-operation, gas turbines with higher operating temperatures, longer life, or both, and, perhaps, lower weight seem within the realm of not-too-distant possibilities.



Official Department of Defense Photo

This diagram depicts the new supersonic Ram-Jet Test Chamber developed for the Air Force by the Wright Aeronautical Corporation of Wood-Ridge, New Jersey. In the new test facility, the Ram-Jet Engine is placed in the center of the 12-by-96-foot stainless steel chamber (cutaway) where it can be subjected to simulated speeds up to 2600 miles per hour, and simulated altitude conditions up to 80,000 feet. Development of the new Ram-Jet Test Chamber will enable the Air Force to collect heretofore unobtainable information on the capabilities and applicabilities of the Ram-Jet Type Engine.

TRANSITION TEMPERATURE OF SHIP PLATE IN NOTCH-TENSILE TESTS

ACKNOWLEDGMENT

The *Welding Journal* for April, 1950 contained this article by Messrs E. M. MACCUTCHEON, C. L. PITTIGLIO and R. H. RARING. The first two were connected with the David Taylor Model Basin and MR. RARING with the National Research Council when this investigation was conducted. It is a study of steels under new specifications used in ship construction together with some experimental steels and an evaluation of specimens used in determining notch toughness and transition temperatures of steel for ship construction.

INTRODUCTION

During the past few years, the Ship Structure Committee has sponsored a series of research projects in which notched, full-thickness specimens of ship plate were tested in tension.[1–6] One of the principal goals of these efforts was to determine the minimum specimen width which would simulate the brittle-to-ductile transition characteristics of a "wide" specimen or, by inference, a welded ship. In these investigations, specimens whose widths ranged from 3 to 108 in. were tested, and it was concluded that a 12-in.-wide specimen would provide a reliable index to the ductility-transition behavior of large plates. Consequently, the 12-in.-wide, notch-tensile specimen has become a standard of reference in the research program of the Ship Structure Committee and currently is being used at Swarthmore College[7] and at the David Taylor Model Basin for tests to evaluate the notch ductility of ship steel.

However, since the 12-in.-wide specimen is an expensive one and is too large to be broken in the testing machines available in most laboratories, it cannot be used for routine testing. It is desirable, therefore, to establish the correlation between results from tests of this 12-in.-wide reference specimen and results from tests of the various types of small specimens; to date, however, insufficient data exist to establish such correlation with certainty.

The work reported in the present paper was planned with cognizance of this history of the 12-in.-wide specimen and is part of the program of the wide-plate tests sponsored by the Ship Structure Committee.

OBJECTIVE AND SCOPE

The objectives of this investigation were:

1. To increase the volume of data from tests of the 12-in.-wide specimen, which data are useful for correlation with results of tests of various small specimens. To this end, six steels conforming to the old A.B.S. (American Bureau of Shipping) specification for hull steel, three steels conforming to the new A.B.S. specification for hull steel (Class B), and two modified A.B.S. Class B steels were tested.

2. To determine the transition temperature of steels removed from two fractured ships; in this phase of the work 12-in.-wide specimens from six plates were tested.

3. To determine the effect on transition temperature, as determined by the 12-in.-wide specimen, of adding aluminum to two otherwise typical A.B.S. Class B heats of steel.

4. To develop a small, economical tensile specimen for determining transition temperature of steel; eleven of the steels which had been tested with the 12-in.-wide specimen were used for this purpose.

MATERIALS

Plate from Pierre S. Dupont

The Liberty Ship *Pierre S. Dupont* suffered several serious cracks of the brittle, cleavage type, while weathering a severe winter storm at sea on Feb. 10, 1948. When this casualty occurred, the air temperature was 27° F. and the water temperature was 42° F.

One of the deck plates (C-10 on the port side) in way of a fracture; but which included neither the source nor the termination of the fracture, was obtained for investigation. Seven 12-in.-wide specimens from this plate were tested; Figs. 1 and 2 illustrate the positions of these specimens with respect to the fracture. Specimens from this plate are designated in this report by the letters "DU."

Plates from Ponaganset

The T-2 Tank ship *Ponaganset* broke completely in two on Dec. 9, 1947, while moored and under a relatively heavy hogging load. The fracture, which occurred at approximately amidship, was of the brittle cleavage type. At the time of failure, the air temperature and water temperature were 34 and 41° F., respectively.

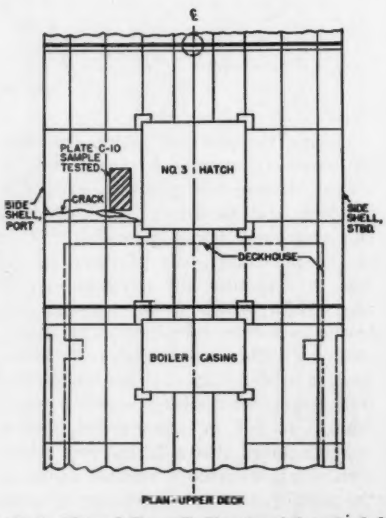


Fig. 1 "Pierre S. Dupont"—Plan view of the upper deck showing the plate removed for testing and the adjacent crack

The crack started in a weld near the corner of No. 3 hatch and was one of several major cracks involved in the casualty

Samples from five of the plates through which the fracture passed were tested using the 12-in.-wide specimen; Figs. 3 and 4 illustrate the positions of these

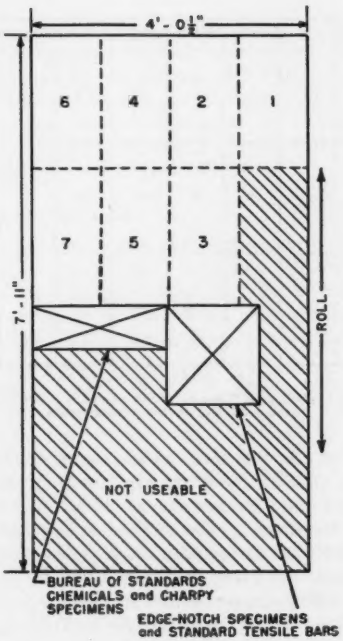


Fig. 2 "Pierre S. Dupont"—Sample from deck plate C-10 showing location of test specimens

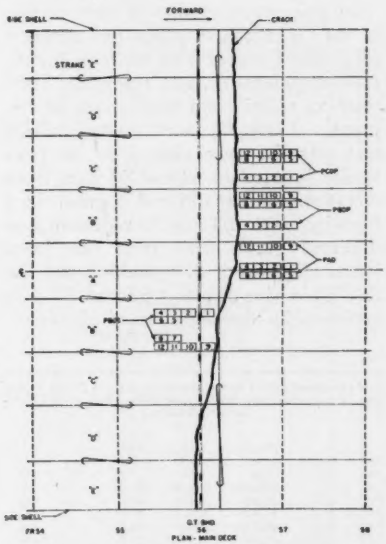


Fig. 3 "Ponaganset"—Plan view of main deck showing the location of test specimens

The crack started in the overlap of two fillet welds around appendages on the main deck. The starting point is about 1 ft. from the starboard side which is at the bottom of the illustration

plates in relation to the path of the crack. The stringer plate, in which the fracture started, was not tested with the 12-in.-wide specimen because there was not sufficient steel available.

In this report the five *Ponaganset* plates are identified by a four-letter code, the first letter of which is "P" (for *Ponaganset*), the second letter indicates the strake, the third letter is either "D" or "S," for "deck" or "side," and the last letter is either "P" or "S" for "port" or "starboard."

Mill Steels

Shortly after the promulgation of the current American Bureau of Shipping specification for ship plate, the Ship Structure Committee procured samples of ¾-in. plates, some made to the old specification and others to the new specification for Class B material. These plates, which are identified by code numbers S1 to S23 and which represent 16 different heats, were obtained for tests to compare the quality of the steel made to the new specification with that made to the old one. In addition to the tests of the mill steels reported here, other steels from this series are being tested at the New York Naval Shipyard and at Swarthmore College.

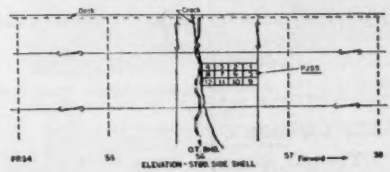


Fig. 4 "Ponaganset"—Elevation of the starboard side-shell showing the location of test specimens in the topside strake

The crack started near the starboard side of the ship in the main deck, which is at the top of the illustration

In this paper, results of tests of five of the "S" series of plates (S1 through S5), which conform to the new A.B.S. class requirements, are reported. They were all rolled from middle cuts of the ingots. Plate S1 was coarse-grained and was the only plate from its particular heat; plate S2 and S3 were from one heat, but they differed in grain size; likewise, plates S4 and S5 represent two different grain sizes from one heat. These steels were all semikilled, and the fine grain size of plates S3 and S4 was achieved by the addition of aluminum

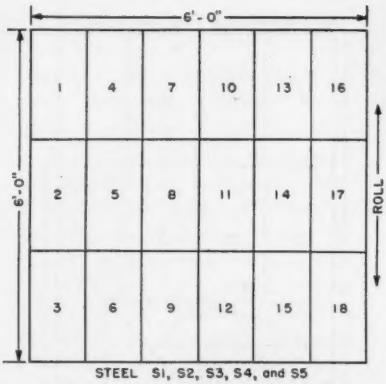


Fig. 5 Mill-Steel Plates S1, S2, S3, S4 and S5 showing the arrangement of 12-in.-wide specimens

(1½ lb. per ton) to the ingot molds. In this paper, grain size refers to the austenitic grain size at 1700° F., as determined by the method of McQuaid and Ehn. Particulars of all of the plates tested are given in Tables 1 and 2.

The locations of the specimens in the plates of S1, S2, S3, S4 and S5 are shown in Fig. 5.

Table 2—Mechanical Properties of Steel from *Pierre S. Dupont, Ponaganset* and Mill Heats (1-In. Wide Full-Thickness Specimens)

Specimen	Gage, in.	Tensile strength, psi.	Yield point, psi.	Elongation in 8 in., %	RA, %	Bend test
DU	3/4	60,200	34,400	28.2	..	
PBDP	13/16	52,300	31,900	32.5	..	OK
PBDS	13/16	55,600	32,600	31.0	..	OK
PAD	13/16	59,700	30,500	25.0	..	OK
PCDP	13/16	54,500	34,500	28.0	..	OK
PJSS	3/4	65,200	37,100	27.5	..	OK
S1	3/4	61,400	37,200	27.8	60.0	
S2	3/4	58,900	30,500	34.0	..	
S3	3/4	58,000	29,200	33.0	61.0	
S4	3/4	65,800	35,900	29.0	58.0	
S5	3/4	65,600	38,000	31.0	61.0	
Dn	3/4	67,200	40,500	27.0	50.1	
E	3/4	62,500	38,000	28.0	52.3	

Table 1—Chemical Analysis in % of Specimens from *Pierre S. Dupont, Ponaganset* and Mill Heats

Plate	C	Mn	Si	P	S	Cu	N	Type steel*	Grain size†
DU	0.26	0.33	<0.02	0.011	0.038	..	..	..	..
PBDP	0.14	0.38	0.00	0.020	0.035	0.16	0.005	Rim	..
PBDS	0.15	0.40	0.00	0.020	0.030	0.17	0.004	Rim	..
PAD	0.22	0.47	0.04	0.021	0.035	0.03	0.004	S-K	..
PCDP	0.14	0.41	0.01	0.027	0.030	0.10	0.004	Rim	..
PJSS	0.29	0.45	0.10	0.015	0.040	0.03	0.0045	S-K	..
S1	0.19	0.67	0.095	0.007	0.037	..	0.0047	S-K	2–4
S2	0.18	0.61	0.055	0.007	0.021	0.05	0.0052	S-K	1–4
S3	0.17	0.61	0.054	0.007	0.020	0.05	0.0046	S-K‡	5–8
S4	0.20	0.92	0.008	0.015	0.040	0.13	0.0038	S-K‡	5–7
S5	0.21	0.90	0.09	0.016	0.040	0.12	0.005	S-K	2–4
Dn	0.22	0.55	0.21	0.013	0.024	0.22	0.004	Si-K	..
E	0.20	0.33	0.01	0.013	0.020	0.18	0.002	Rim	..

* Rimmed, semikilled (S-K) or Si-killed (Si-K).
† McQuaid-Ehn.
‡ The steel for these plates was modified by the addition of aluminum in the ingot mold.

CONDITIONS OF TEST FOR 12-IN.-WIDE SPECIMENS

Test Specimen

The test specimen was a 12- x 24-in., full-thickness, notched plate; this plate was loaded by tension in the direction of the longest dimension, which was the direction of rolling. The notch effect was introduced by a 3-in.-long slot perpendicular to the direction of loading and terminating in jeweler's hack-saw

cuts approximately 0.012 in. in width. This specimen has been used at the University of California, University of Illinois, Swarthmore College and for previous investigations at the David Taylor Model Basin. Particulars of the specimen are given in Fig. 6.

Instrumentation

The apparatus for measurement of strain was modeled after that developed at Swarthmore College for similar tests. Twelve spring clips fitted with bakelite-bonded SR-4 strain gauges were used, five on each of the specimen's faces and one on each edge. The strain gauges were connected by a circuit that averaged the results from all gauges on a single strain indicator. Figure 7 shows the gauges mounted on a specimen.

The spring clips were mounted on a 9-in.-gauge length, which was equal to ¾ of the width of the specimen. Before each series of tests was started, the strain-gauge assembly was calibrated with a dial micrometer.

A mechanical extensometer was rigged between the heads of the specimen to provide elongation data which could be used in the event of failure of the SR-4 gauges; no such failures occurred, however, and all the elongation data presented were taken from the wire gauges.

Temperature Control

The temperature-control equipment consisted of an insulated storage box for dry ice, a blower and a jacket around the specimen, as shown by Fig. 8. Electric-resistance heaters were located inside the jacket for tests above room temperature. The blower and heater were automatically controlled by a thermostat placed inside the specimen jacket. The exact temperature of the specimen was indicated by two thermocouples embedded in the specimen close to the ends of the slot. Silica gel was placed inside of the dry-ice storage chest to reduce the moisture content of the circulating air and thus prevent corrosion of the equipment and electrical difficulties with the strain gauges.

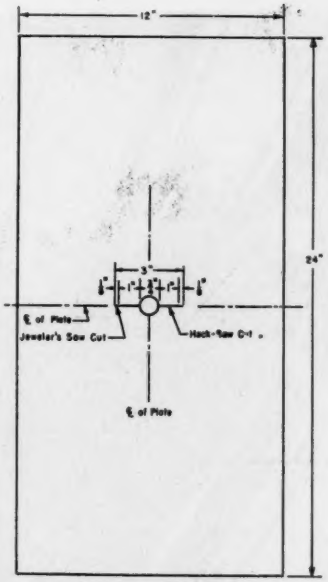


Fig. 6 Twelve-inch-wide-specimen dimensions
Both the direction of rolling and the direction of pulling corresponded to the 24 in. dimension, i.e., perpendicular to the saw-cut notches

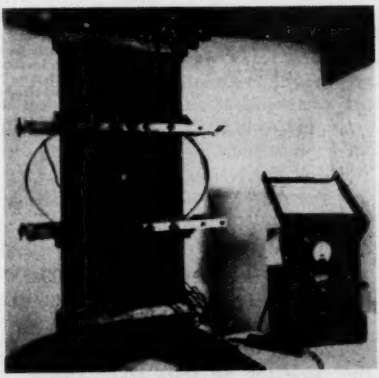


Fig. 7 Twelve-inch-wide specimen welded to pulling headers and set in the grips of the 600,000-lb. testing machine
The twelve 9-in. baselength clip gages are in place. The small triangular contact panel in front of the strain indicator was used to facilitate the detection of defective strain gages

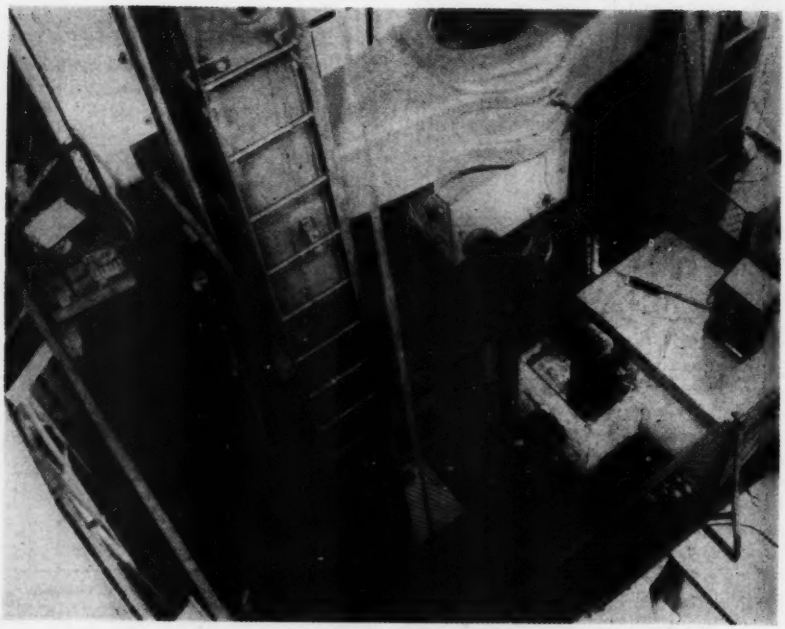


Fig. 8 Twelve-inch-wide-specimen temperature-control equipment

The ice box is on the right under the strain indicator. The motor in the right foreground powers the fan which is buried in a box of sawdust, from the top of which the heavily lagged supply duct goes to the box-shaped jacket around the specimen. On the left side of the jacket the fan-control thermocouple can be seen and to the right the return duct to the ice box

Testing Technique

The tests were performed in a 600,000-lb. universal testing machine at the David Taylor Model Basin.

The rate of application of load was the same for all tests and followed a fixed schedule. The load was increased from zero to yield at the rate of 50,000 lb. a minute; above the yield load, the rate of application of the load was regulated on the basis of the elongation in the 9-in. gauge length, as indicated in the table.

Elongation	*Controlled rate of elongation*
From yield to 0.2 in.	0.01 in./min.
From 0.20 to 0.40 in.	0.02 in./min.
From 0.40 in. to failure	0.05 in./min.

Elongation readings were made at 1-min. intervals until the yield load was passed; after yielding, readings were made at 2-min. intervals.

It was not possible, due to character-

istics of the apparatus, to detect the occurrence of the first crack. The small window in the temperature-control jacket did not allow a sufficiently good view of the notch root for observation of the initiation of the crack, and the method of connecting the individual gauges had the effect of masking any manifestation of crack initiation by means of the elongation measurement.

Load-elongation curves were drawn for each of the specimens tested, and the area under these curves was measured with a planimeter. From these areas, "energy to maximum load" and "energy to rupture" were determined. Figure 9 shows a typical example of such a curve for a ductile specimen.

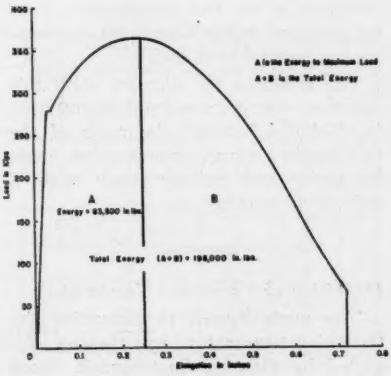


Fig. 9 Twelve-inch-wide specimen—Typical load-elongation diagram for a ductile specimen

The surfaces of the fractures of all specimens were examined, and the amount of fibrous and crystalline area was estimated. Thus the transition temperature could be based on either absorbed-energy or fracture-mode criteria.

1-IN. EDGE-NOTCH TEST

In using the relatively large and costly 12-in.-wide specimen, it is obviously desirable to know the approximate transition temperature before the temperatures for the first few tests are selected. In the investigation of steels from fractured ships this is especially important because frequently the amount of available material limits the number of large specimens that can be manufactured. An effort therefore was made to estimate the transition temperature of the large specimens from results of tests of small specimens.

In this investigation certain economic considerations and mechanical difficulties made the use of any of the more conventional small specimens undesirable. A small edge-notch tensile specimen, therefore, was devised that could be prepared by unskilled personnel and without the use of machine tools other than a power saw and a drill press, and which could be tested in a testing machine of 30,000-lb. capacity. Small edge-notch tensile tests are, of course, not novel; Davis, *et al.*,[4] Wagner and Klier,[8] and Tipper[9] have used such tests.

Experimentation was performed with a 1-in.-wide, full-thickness specimen with notch geometries as illustrated by Fig. 10. To achieve control of temperature in the testing of these specimens it was found to be satisfactory to cool the specimens in a suitable bath to some temperature 10 to 20° F. below the desired test temperature and then quickly to transfer the specimen to the testing position in the machine. Thermocouples were embedded in the specimen in the vicinity of the notch root, and the temperature was observed during the course of the test.

The speed of testing was so controlled that failure occurred after approximately 2 min. The temperature of the specimen 60 sec. after the start of the test, at which time the load was 15,000 lb. and appreciable yielding had occurred, was

recorded as the test temperature. This temperature differed from the temperature at rupture by 3 to 5° F.

The criterion for ductility with this specimen was the maximum contraction in thickness between the roots of the two notches, a measurement that could be easily and quickly made with a pointed micrometer.

RESULTS

Definition of Transition Temperature

The method used to determine the transition temperature was the one suggested by Osborn, Scotchbrook, Stout and Johnston,[10] in which the observed values of the criterion used for determining the transition temperature were averaged for each temperature and the plotted points so obtained were joined by a series of straight lines. The transition temperature was defined as the point on the curve corresponding to the value of the criterion midway between its maximum and minimum values.

12-In.-Wide Specimens

The transition temperatures of the steels, as determined by tests of the 12-in.-wide specimens and according to the criteria, total absorbed energy and fracture appearance, are listed in Table 3. Both stress, in pounds per square inch, and energy, in inch-pounds per square inch, are based on the net effective area of the cross section of the specimen in way of the slot. Data from the individual tests are presented graphically by Figs. 11 and 12.

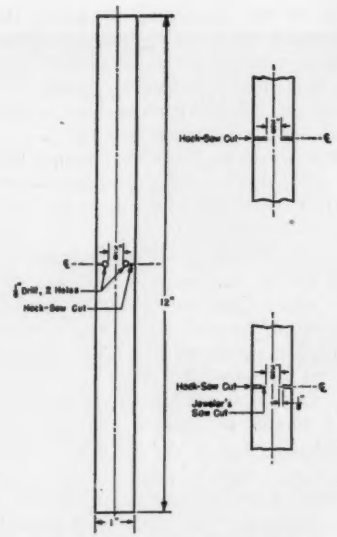


Fig. 10 Dimensions of 1-in. edge-notch specimen
In all of the tests reported the specimens were oriented so that the long dimension corresponded to the direction of rolling of the plate

1-In. Edge-Notch Specimens

Data from tests of the 1-in. edge-notch specimens are presented graphically by Figs. 13, 14 and 15. Numerals opposite plotted points on these figures designate duplicate results. Table 3 lists the transition temperatures of the steels as determined by this specimen, on the basis of the contraction at the root of the notch.

Table 3—Transition Temperatures in° F. of Steels as Determined by Various Tests

	12-in.-wide-specimen tests			*1-in. edge-notch tests*					
Steel	*No. tests*	*T. T., Energy to rupture*	*T. T., fracture appearance*	*1/8 in., hole*	*Hack saw*	*Jeweler's hack saw*	*Keyhole Charpy, 15 ft.-lb.*	*V-Notch Charpy, 15 ft.-lb.*	*Navy Tear test*
DU	7	37	37	..	107	..	..	62	..
PBDP	11	46	46	71	..	..	9	..	90
PBDS	11	60	60	127	..	..	18	..	110
PAD	11	104	102	152	..	..	12	..	150
PCDP	11	54	55	54	..	..	−2	..	90
PJSS	11	58	60	89	..	..	−6	..	90
S1	12	54	50	73	118	130	15*	..	100
S2	15	61	62	..	124	..	8*	..	110
S3	15	55	56	..	195	..	6*	..	80
S4	15	−27	−14	..	31	..	−50*	..	−20
S5	15	31	34	..	76	..	−5*	..	70
Dn	28†	18	19	40*	87*	72*	−62*	−18*	70*
E	35†	81	77	160*	180*	175*	27*	90*	140*

* Not the same plate as used for the 12-in.-wide specimens.
† Tests by Swarthmore College and the University of Illinois.

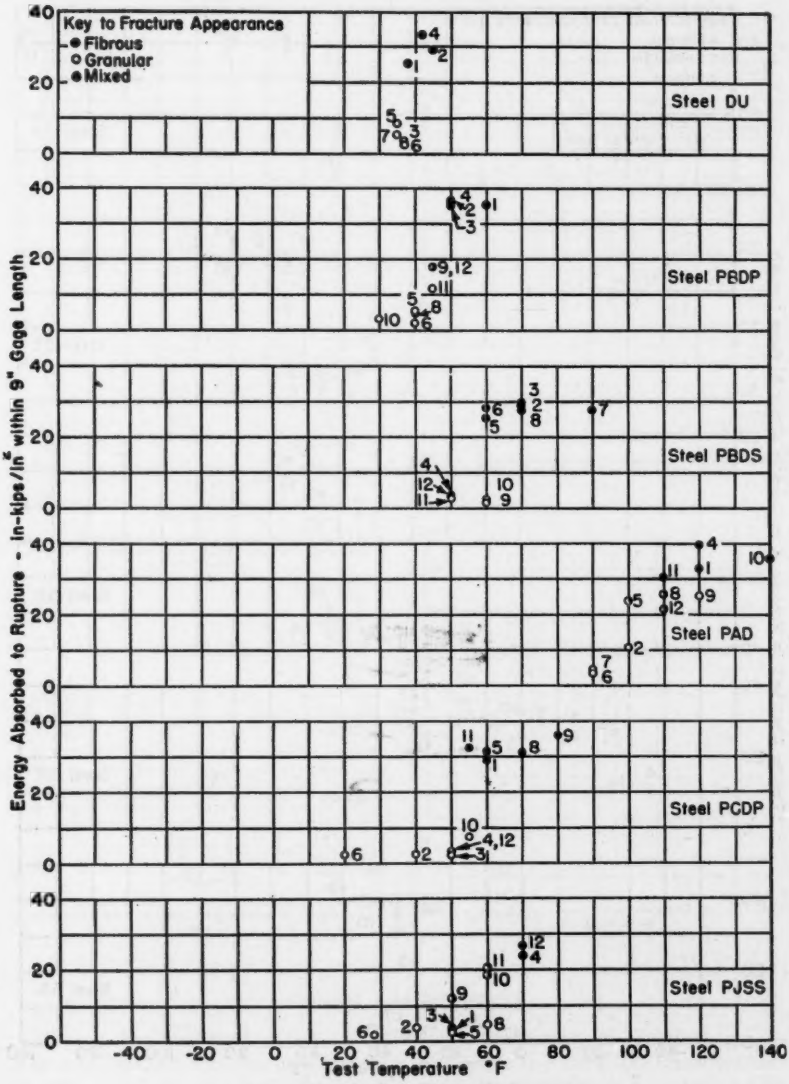


Fig. 11 Twelve-inch-wide specimens—Results of tests of samples removed from fractured ships

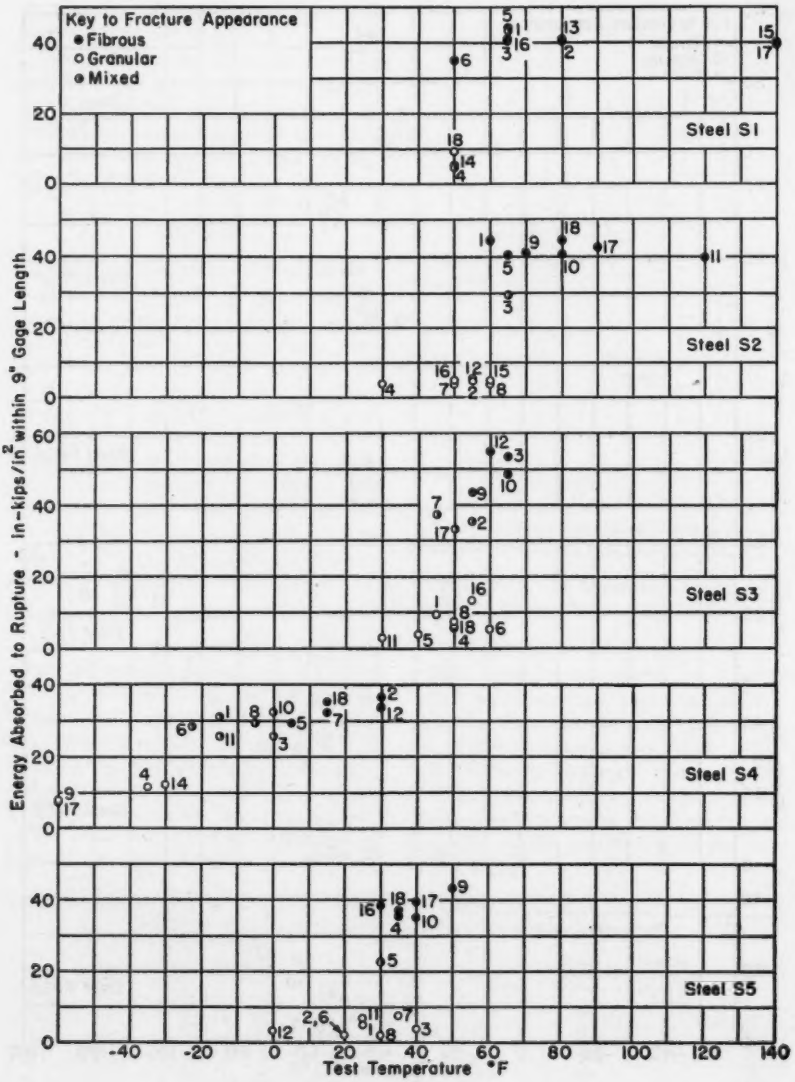


Fig. 12 Twelve-inch-wide specimens—Results of tests of mill steels

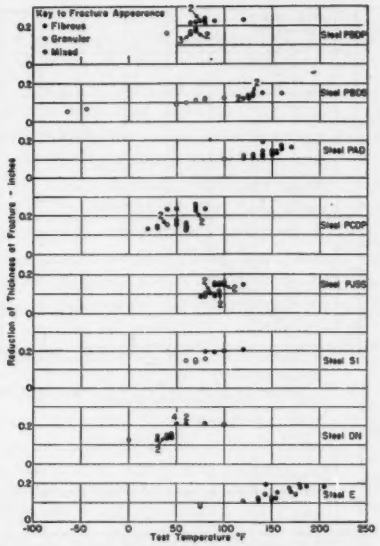


Fig. 13 One-inch edge-notch specimen—Results of tests of specimens with notches terminating in a 1/8-in. diameter hole

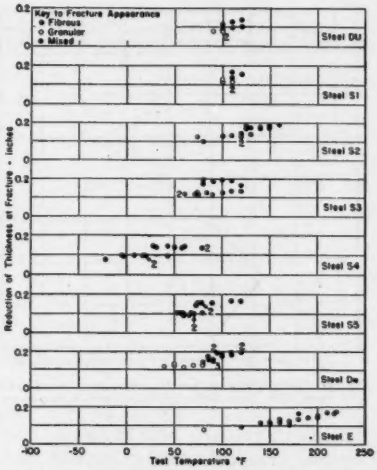


Fig. 14 One-inch edge-notch specimen—Results of tests of specimens with notches terminating in ordinary saw-cuts

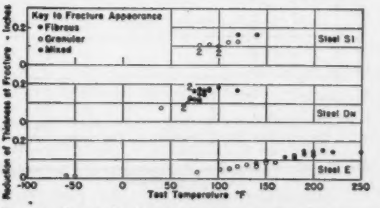


Fig. 15 One-inch edge-notch specimen—Results of tests of specimens with notches terminating in jeweler's saw cuts

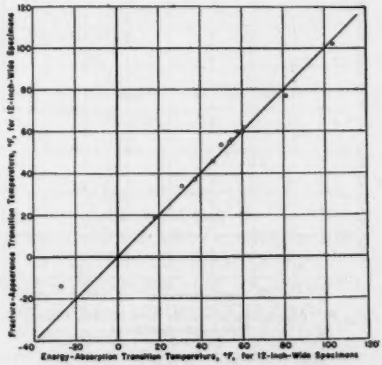

Fracture-Appearance Transition Temperature, °F, for 12-Inch-Wide Specimens
Energy-Absorption Transition Temperature, °F, for 12-Inch-Wide Specimens

Fig. 16 Correlation between transition temperatures by fracture appearance and total absorbed energy, 12-in.-wide specimen

FINDINGS AND DISCUSSION

One of the principal objectives of this investigation was the development of data useful for the establishment of correlation between test results from the 12-in.-wide specimen and those from the various small specimens. This correlation is graphically represented by Figs. 16 through 20. In these graphs, each point represents the transition temperature as determined by each of the two methods being compared. The ΔT indicated is the arithmetic mean of the difference between the transition temperatures of each steel as determined

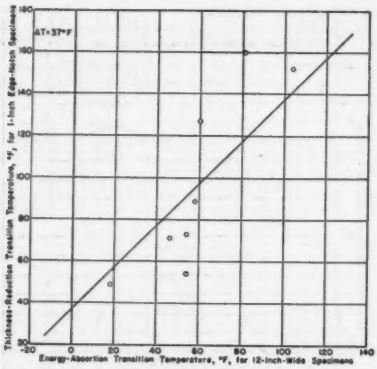

ΔT=37°F
Thickness-Reduction Transition Temperature, °F, for 1-Inch Edge-Notch Specimens
Energy-Absorption Transition Temperature, °F, for 12-Inch-Wide Specimens

Fig. 17 Correlation of results from 1-in. edge-notch specimens with $\frac{1}{2}$-in. holes and 12-in.-wide specimens

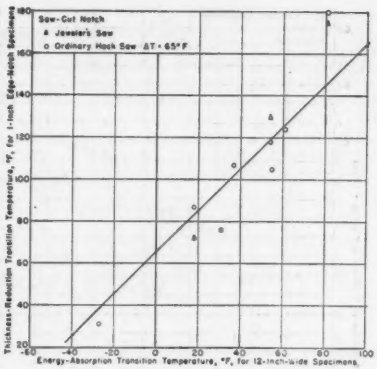

Saw-Cut Notch
▲ Jeweler's Saw
○ Ordinary Hack Saw ΔT = 65°F
Thickness-Reduction Transition Temperature, °F, for 1-Inch Edge-Notch Specimens
Energy-Absorption Transition Temperature, °F, for 12-Inch-Wide Specimens

Fig. 18 Correlation of results from 1-in. edge-notch and 12-in.-wide specimens

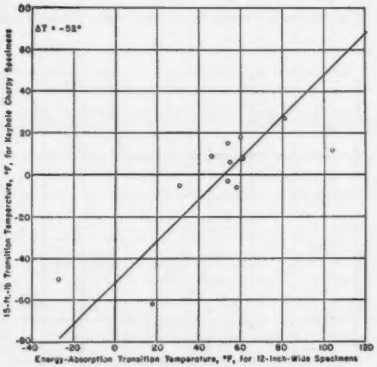

ΔT = -52°
15-Ft-lb Transition Temperature, °F, for Keyhole Charpy Specimens
Energy-Absorption Transition Temperature, °F, for 12-Inch-Wide Specimens

Fig. 19 Correlation of results from keyhole Charpy and 12-in.-wide specimens

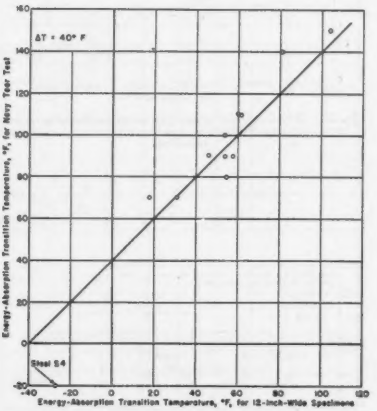

ΔT = 40° F
Steel S4
Energy-Absorption Transition Temperature, °F, for Navy Tear Test
Energy-Absorption Transition Temperature, °F, for 12-Inch-Wide Specimens

Fig. 20 Correlation of results from Navy tear test and 12-in.-wide specimens

by the methods being compared. The straight lines are drawn at 45° through the value of ΔT. This construction represents a simple, though not necessarily true, relationship between the tests.

For the edge-notch tensile specimens with saw-cut notches, the value of ΔT is based on those specimens with ordinary hack-saw cuts only. The specimens notched with a jeweler's hack saw, however, showed transition temperatures so close to those with the ordinary hack-saw cut that both types are plotted on the same graph.

As is evident from Fig. 16, transition temperature on the basis of fracture appearance shows excellent correlation with transition temperature on the basis of absorbed energy for the 12-in.-wide specimen.

Correlation between transition temperatures as determined by the edge-notch tests, the Charpy tests or the

Navy tear test on the one hand, and tests of the 12-in.-wide specimens on the other, is poor to fair, as indicated in Table 4, and it must be concluded that precise relationships between the tests are not established.

In comparing results from the Navy tear test and the 12-in.-wide specimen, it should be noted that the definitions for transition temperature differed in the two tests; in the Navy tear test, transition temperature is defined as the highest temperature at which one or more specimens showed an energy-to-propagate value of less than one-half of the energy level of the predominantly shear region, while the transition temperature with the 12-in.-wide specimen was as defined under "Results." However, when the transition temperature by the Navy tear test was determined by using the same definition as was used for the 12-in.-wide specimen, it was found that the scatter in the correlation between results from the Navy tear test and the 12-in.-wide specimen was essentially unchanged and the ΔT was increased by only about 4.5° F. Consequently, the commonly used definition for transition temperature for the Navy tear test was used throughout this paper.

Effect of Addition of Aluminum

It is interesting to note from Table 3 that the addition of aluminum to steel S2 lowered the energy-transition temperature only about 6° F., while the same addition lowered the transition temperature of steel S5 by about 58° F. The grain-refinement effect however, was about the same for both steels, reducing the grain size by 5 or 6 numbers. A very likely explanation for this difference is the observed occurrence of fissuring in steel S4. Thus, in the case of steels S5 and S4, which were from the same heat and differed only in that aluminum was added to steel S4, steel S5 showed no fissures in the 12-in.-wide specimens while steel S4 showed extensive fissuring. In the case of steels S2 and S3, which also were from one heat but differed in that aluminum was added to steel S3, neither showed significant fissures in the 12-in.-wide specimens. The effect of fissuring in lowering transition temperature has been observed previously.[4]

Table 4—Correlation of Transition Temperatures with 12-In.-Wide Specimen Energy-Absorption Transition Temperature

Test specimen	*Criterion*	*No. of steels*	*ΔT, ° F.*	*Deviation, ° F. Maximum*	*Mean*
12-in.-wide Specimen	Appearance	13	1	12	2½
1-in. edge-notch, ½-in. hole	Thickness reduction	8	38	41½	20½
1-in. edge-notch HSC	Thickness reduction	8	65	34	11
1-in. edge-notch JSC	Thickness reduction	3	75	20½	14
Keyhole Charpy	15 ft.-lb.	12	−52	40	14½
V-Notch Charpy	15 ft.-lb.	3	−1	35	23½
Navy tear test	Energy	12	40	31½	10½

Transition Temperature of Steels Tested with the 12-In.-Wide Specimens

No conclusion as to the role of the steel quality in the failure of the *Pierre S. Dupont* and *Ponaganset* can be justified on the basis of the results in Table 3. While it is true that plate PAD from *Ponaganset* had a relatively high transition temperature, it should be realized that the fracture did not start in this plate but merely traversed it, as it did other plates with lower transition temperatures, e.g., plate PBDP which had a transition temperature 58° F. lower than that of plate PAD.

Likewise, the results of the tests of the ABS Class B steels are insufficient to permit any generalization as to the quality of steels made to the new Class B specification. Judgment of the merits of the specification revision represented by the Class B steel must be based on a broader sampling and further tests.

1-In. Edge-Notch Tension Test

In the course of testing the 1-in. edge-notch tensile specimens, it was observed that those specimens which were covered with mill scale frequently revealed yielding by the sudden separation of scale flakes. This yielding was studied in several specimens by means of a lateral-contraction gauge located at the notch root in order to determine how the nominal stress at yielding in the edge-notch specimens compared with the yield point as determined by conventional unnotched tensile specimens. Results of these preliminary studies showed that while some edge-notch specimens had a sharply defined yielding, other specimens did not, and a definite yield phenomenon in the edge-notch specimens could not be correlated with the yield point as determined by unnotched specimens. These preliminary studies were therefore abandoned.

Correlation with results from tests of the 12-in.-wide specimen indicates that the edge-notch specimen with a saw-cut notch can predict transition behavior in the large specimen at least as well as other small specimens.

CONCLUSIONS

Transition temperature as determined by the 12-in.-wide specimen is essentially the same by either the absorbed-energy or the fracture-appearance criterion.

Since this investigation represents only one phase in a series of wide-plate tests, no other general conclusions are warranted on the basis of the data presented in this paper.

ACKNOWLEDGMENT

A considerable amount of the data used for correlation of test results was supplied by N. A. Kahn of the New York Naval Shipyard, J. L. Wilson of the American Bureau of Shipping, and Dr. Morgan L. Williams of the National Bureau of Standards; certain of these data were received in private communications. Permission to use data from all of the above sources is gratefully acknowledged. Completion of the tests could not have been accomplished without the assistance of P. E. Humphries and E. H. Boblits of the David Taylor Model Basin. The plates from the *Ponaganset* were secured through the cooperation of the Boston Naval Shipyard.

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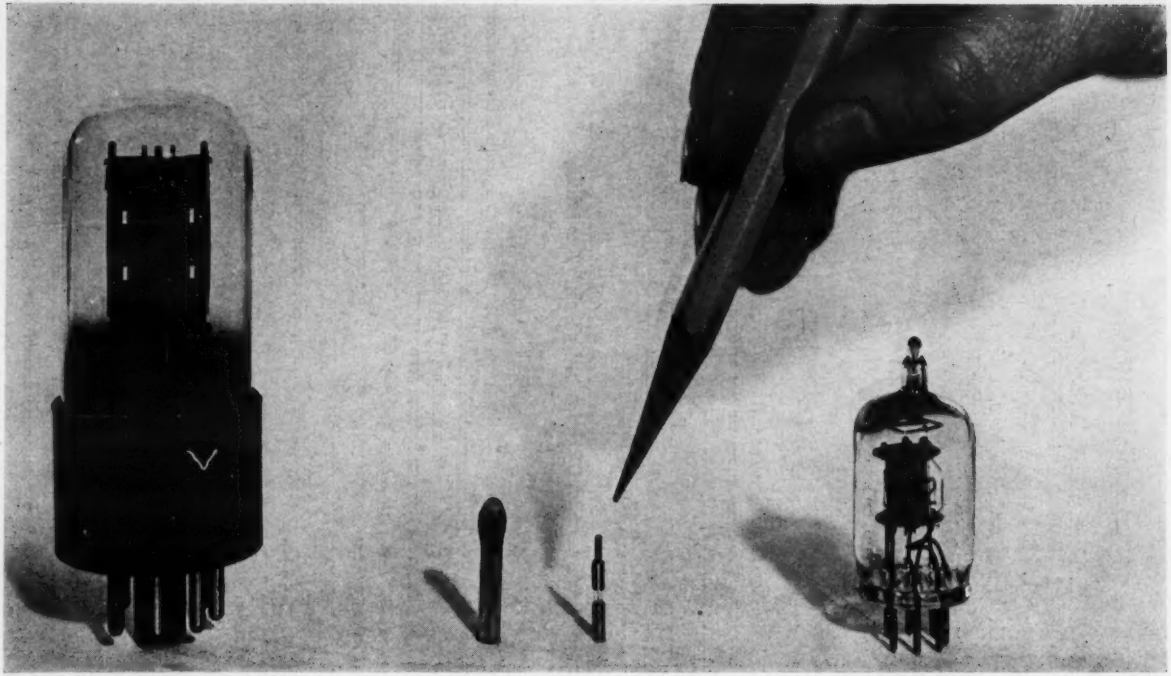


Official Department of Defense Photo

A new, sub-miniature radio tube, 1/90th the size of the present-day tube, is now in the early stage of development at the Air Materiel Command's Components, and System Laboratory at Wright-Patterson Air Force Base, Dayton, Ohio. This close-up shows the comparative size of the miniature tube with two conventional tubes, a pencil, and match head.

SCREWSHAFT CASUALTIES—THE INFLUENCE OF TORSIONAL VIBRATION AND PROPELLER IMMERSION

ACKNOWLEDGMENT

The January, 1950 Transactions of the Institute of Marine Engineers contained this article by MR. S. ARCHER, Senior Engineer Surveyor for Research, Lloyd's Register of Shipping. The paper describes investigations carried out during the past two years by that Research Staff with a view to determining the propeller factors underlining the many screw shaft failures in the Liberty vessel. The complete article included several appendices which are listed but not reprinted at the end of this reprint.

INTRODUCTION

Soon after the conclusion of the late war a pronounced increase in the screwshaft casualty rate of certain classes of 10,000-ton dead weight merchant vessels, built in America under the emergency program, began to cause concern in maritime circles, at first mainly in America, but later also in Europe and elsewhere, as transfers of the U.S. surplus fleet to other flags began to take effect. The casualties took the form of fatigue cracks, or fractures, at the large end of the screwshaft cone.

GENERAL

The class of vessel chiefly affected has been the well-known "Liberty," or E.C.2-S.C.1 type, although the trouble has also been common in the Canadian "Victory" type and, to a lesser extent, in some of the oil-burning vessels of the "Canadian" type.

The approximate general specification of these vessels is as follows:—length 416 feet b.p., breadth 56.9 feet moulded, depth 37.33 feet moulded to upper deck, summer draught 27 ft., 9 in., gross tonnage about 7160, dead weight tonnage 10,000.

Fig. 1 shows the general arrangement of the "Liberty" type vessel, including bunkering and ballasting capacities.

It may be stated that all these classes are generally similar as regards hull form and propelling machinery and

comparative size of the miniature tube with two conventional tubes, a pencil, and match head.

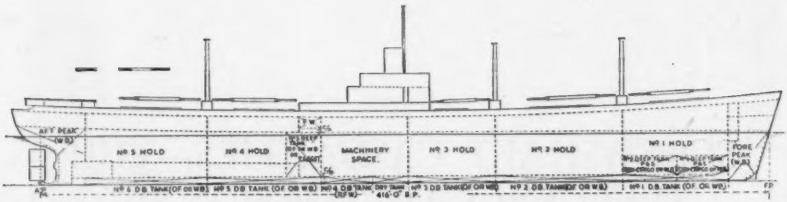


FIG. 1—*Ballast and bunker arrangements for "Liberty" vessels*

differ in the former respect only slightly from the well-known U.S. built "Ocean" type, which, in turn, was based on the British "North Sands" design developed on the North-East coast, the prototype vessel being named S.S. *Empire Liberty*.

The principal differences between the various American-built vessels lie in the steam generating plant, in the superstructure, and extent to which welding is employed, and in the bunkering and ballasting arrangements.

The propelling machinery of the U.S. built "Oceans" and "Libertys" and the Canadian "Victorys" consists of a triple-expansion steam engine having three cylinders, 24½-inch, 37-inch, and 70-inch diameter by 48-inch stroke, with a steam pressure of 220 lb. per sq. in., and with superheat. The design was developed by The North-Eastern Marine Engineering Co., Ltd. The shafting sizes and arrangement, also the propellers, stern frames, and rudders are for all practical purposes identical (see Figs. 2 and 13). The rated service horsepower of the main engines is 2500 Ihp. at 76 Rpm., and details of the 4-bladed, solid, manganese bronze propeller are indicated in Fig. 2. The rudder is of a special American design, known as the "Goldschmidt" type, whereas in the original "North Sands" class a conventional semi-balanced type was fitted.

The principal differences between the types are that whereas the "Ocean" vessels have coal-fired Scotch boilers of a total heating surface of 7140 sq. ft., both the "Libertys" and the Canadian "Victorys" are oil-burning with water-tube boilers of approximately 10,000 sq. ft. heating surface. The "Canadians" have Scotch boilers, either oil- or coal-fired. Further, the "Libertys" and a few of the Canadian "Victorys" have deep tanks of a total capacity of 650 tons in No. 1 hold, in addition to the deep tank abaft the engine-room of capacity 760 tons. The latter is also fitted in the "Oceans," which have, however, a 500-ton cross-bunker forward of the boiler-room, and a 600-ton 2nd deck bunker in way of the engine- and boiler-rooms. The presence of the cross-bunker, of course, cuts down the effective length of No. 3 hold, but on the other hand, the latter may be used as additional bunker space (955 tons) on long voyages.

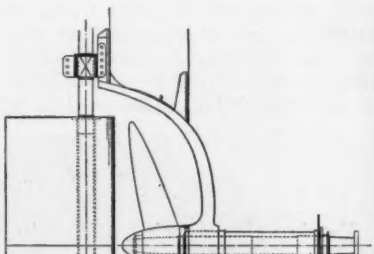


FIG. 2—*Stern arrangement of "Liberty" ship*

Propeller data:
Diameter, 18ft. 6in.
Pitch varying (18ft. mean)
Surface, 117 sq. ft.
No. of blades, 4 r.h.
Material manganese bronze
Boss/diameter ratio 0·14

STATISTICS

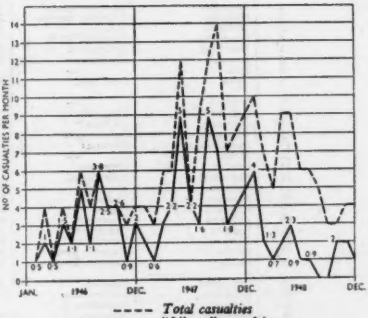
The construction of the largely welded dry cargo "Liberty" type of vessel was begun in the United States in January 1942, and altogether no less than 2580 were built, of which 2315 remained at the conclusion of hostilities, representing a total gross tonnage of nearly 17 million tons, or about 20 percent of the world total for 1947 (13½ percent in number) of all vessels greater than 500 tons gross.

Information at 1st December 1948, revealed that altogether a total of 583 "Liberty" screwshafts have been renewed, including during the past three years about 100 casualties at sea with resulting loss of propeller. The cost of these breakdowns in salvage and demurrage charges alone needs no emphasis.

Fig. 3 shows the monthly world totals of propeller losses at sea for the three years ending 31st December 1948, as derived from information published in Lloyd's List. The diagram also indicates the monthly proportion of "Liberty" casualties, and in the lower graph the number of non-"Liberty" failures. The figures in brackets on the "Liberty" graph give the estimated casualties per 1000 "Liberty" vessels in service during each month and do not include "Liberty" tankers.

It will be seen that the heaviest casualty rate for "Libertys" occurred in the summer and autumn of 1947 and that the present rate is very considerably reduced.

The large drop occurring in early 1948 is most probably to some extent a reflection of the recommendations of the American Bureau of Shipping, firstly, in May 1947, for more frequent examination of shafts (2 years instead of 3 years), and secondly, in January 1948, for a reduction in maximum engine speed to 66 Rpm. loaded, and 70 Rpm. light. On the other hand, it should be remarked that the smaller number of casualties at sea must inevitably have



Figures on the curve give approximate casualties per 1,000 "Liberty" ships in service (based on United States statistics of number of "laid-up" vessels)

FIG. 3—*Statistical survey of losses of propellers at sea over period January 1946 to December 1948 (based on Lloyd's list)*

been influenced by the very appreciable reduction in numbers of "Liberty" vessels in service between the peak casualty period of July 1947, and the spring of 1948, say 20 to 25 percent fewer vessels at sea. Accordingly, in the absence of parallel information on the rate of renewal of "Liberty" screwshafts from all causes, it would be premature to assume that all is now well. Meteorological sources state that North Atlantic weather in 1947 was not as a whole unusually stormy, although periods of uncommonly severe gales did occur in the early part of the year, particularly in early January.

Analysis of the statistics of non-"Liberty" propeller losses at sea shows that the ratio of casualties for steam-reciprocating vessels, compared with motor vessels, is approximately the same as that of their respective world totals

TABLE I

Vessel No.	Port of departure	Destination	Ballast or loaded	Leaving draughts		Remarks
				Forward, ft. in.	Aft, ft. in.	
1	Marseilles	Hampton Roads	Troops	—	—	—
2	Colombo	Lourenco Marques	Ballast	—	—	Mean draught 12 feet
3	Bordeaux	U.S.A.	Ballast	11 7	19 11	2,000 tons sand
4	Casablanca	Hampton Roads	—	—	—	—
5	New York	Antofagasta	Ballast	6 0	17 0	—
6	Savona	Charleston	Ballast	—	—	—
7	Split	—	—	—	—	—
8	Havre	New York	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
9	Trinidad	—	—	—	—	—
10	Port Alfred	New York	Ballast	—	—	—
11	Galveston	La Plata	Ballast	9 0	18 0	—
12	Trinidad	Buenos Aires	Ballast	—	—	—
13	New Orleans	Demerara	Ballast	8 7	15 3	—
14	Mobile	—	Ballast	7 2	16 0	—
15	London	Vancouver	Ballast	—	—	Mean draught 12 ft. 8 in.
16	Rotterdam	Philadelphia	Ballast	—	—	Mean draught 13 feet
17	Bordeaux	Montreal	Ballast	7 0	15 0	—
18	Shanghai	Vancouver	Ballast	—	17 6	—
19	Port Said	Naples	Ballast	—	—	—
20	Colombo	Vancouver	Ballast	8 6	16 3	1,170 tons O.F.
21	Leghorn	Charleston	Ballast	—	—	—
22	Glasgow	Vancouver	Ballast	—	—	720 tons O.F.
23	Savona	Hampton Roads	Ballast	—	—	—
24	Rouen	Hampton Roads	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
25	Port Alfred	Paranam	Ballast	—	—	—
26	Hong Kong	San Francisco	Ballast	—	—	1,000 tons O.F.
27	Trinidad	Mackenzie	—	—	—	—
28	Singapore	Vancouver	Ballast	11 8	15 10	Loaded some cargo Port Swets. and Colombo
29	Rouen	New York	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
30	Durban	Buenos Aires	Ballast	11 2	16 0	—
31	Lourenco Marques	Buenos Aires	Ballast	12 6	15 6	—
32	Shanghai	W. C., N. America	Ballast	—	18 0	—
33	Civitavecchia	New Orleans	Ballast	—	—	—
34	Trinidad	Baltimore	Ballast	—	—	—
35	Ancona	Philadelphia	Ballast	—	—	—
36	Hamburg	Key West	Ballast	—	19 0	—
37	Hampton Roads	France or Alex.	Coal	—	—	—
38	Hong Kong	Honolulu	Ballast	—	—	500 tons O.F.
39	Antwerp	Savannah	Ballast	—	14 0	—
40	Durban	Montevideo	Ballast	8 10	15 10	—
41	New York	Antwerp	Part loaded	24 9	28 9	—
42	Venice	Hampton Roads	Ballast	—	—	—
43	Antwerp	Hampton Roads	Ballast	—	14 0	—
44	Liverpool	New York	Part loaded	9 0	17 9	1,166 tons cargo
45	Galveston	Mobile	Ballast	5 6	14 6	—
46	Rouen	Delaware Capes	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
47	Rouen	Hampton Roads	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
48	Yokohama	Balboa	Ballast	—	—	Bunkered Yokohama
49	Genoa	Hampton Roads	Ballast	—	—	—
50	Rouen	Demerara/U.S.A.	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
51	Emden	Montreal	Ballast	—	13 0	—
					16 0	
52	Baltimore	La Guaira	Part loaded	18 10	23 4	—
53	Genoa	Hampton Roads	Ballast	13 3	17 0	—
54	St. Nazaire	New York	Ballast	7 0	14 0	—
55	Gibraltar	Hampton Roads	Ballast	—	—	—
56	Port de Bouc	Costa	Ballast	—	—	—
57	Cork	New York	Ballast	9 3	14 3	—
58	Lisbon	Baltimore	Ballast	—	—	—
59	Hampton Roads	New Orleans	Ballast	—	—	—
60	Lisbon	Hampton Roads	Ballast	—	—	—
61	Rouen	Hampton Roads	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
62	Manila	U.S. Gulf Ports	Ballast	8 9	14 5	—
63	Yokohama	San Francisco	Ballast	—	—	—
64	Trieste	Baltimore	Ballast	—	—	—
65	Aarhus	Philadelphia	Ballast	—	17 9	—
66	Antwerp	New York	Ballast	—	17 8	—
67	New York	Baltimore	Ballast	6 0	13 0	—
68	Venice	United States	Ballast	—	—	—
69	Guam / Honolulu	Balboa	Ballast	9 6	16 2	Bunkered Honolulu
70	Demerara	Trinidad	Part loaded	—	—	3,730 tons cargo 16 ft. 6½ in. mean draught
71	Rouen	Hampton Roads	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
72	Genoa	Hampton Roads	Ballast	7 10	13 0	—
73	Liverpool	Takoradi	Ballast	11 0	17 0	—
74	Rouen	New York	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
75	Trieste	Costa	Ballast	—	—	—
76	Boston	Porto Alegre	General	—	—	—
77	Genoa	New Orleans	Ballast	6 2	17 3	—
78	Norrkoping	Charleston	Ballast	—	18 0	—
79	Avonmouth	Mobile	Ballast	6 4	17 6	—
80	Savona	Mobile	Ballast	—	—	—
81	Buenos Aires	Calcutta	Grain	27 8	29 2	—
82	Gibraltar	Hampton Roads	Ballast	—	—	—
83	Cherbourg	New York	Ballast	5 11	14 5	—
84	Helsinki	Key West	—	—	—	—
85	Marseilles	Hampton Roads	—	—	—	—
86	Rouen	Hampton Roads	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
87	Tampa	Rotterdam	Loaded	26 8	27 11	—
88	Casablanca	U.S.A.	Ballast	—	—	—
89	Rio de Janeiro	Cuba	Ballast	—	—	—
90	Rouen	New York	Ballast	—	—	Between 14 ft. 6 in. and 16 feet mean draught
91	Calcutta	Camden, N. J.	Part loaded	18 7	21 6	—
92	Yokohama / Honolulu	Balboa	Ballast	7 0	15 6	Bunkered Honolulu
93	Marseilles	New York	Ballast	17 8	20 10	—
94	Amsterdam	Hampton Roads	Ballast	10 3	15 0	—
95	Cork	New York	Ballast	8 0	15 2	—
96	Melbourne	Liverpool	Part loaded	22 7	25 3	—
97	Genoa	North of Hatteras	Ballast	—	—	—
98	Galveston	Puerto	Wheat	—	—	—
99	Abadan	New Caledonia	—	—	—	—
100	London	Portland, Or.	—	—	—	—
101	Dairen	Vancouver	Ballast	—	—	—

FIG. 4(a)—*Geographical location in the Atlantic area of "Liberty" ships at the time of propeller loss*

in numbers, thus indicating that neither class of vessel is inherently more prone to screwshaft breakages at sea.

It may be of interest to observe that for the 12-monthly period ending 31st December 1947, of the total of 6161 vessels classed with Lloyd's Register, of which more than 4000 screwshafts were examined by the surveyors, only 13 were found broken and 74 condemned for cracks, representing respectively 0.21 percent and 1.2 percent of the total vessels classed, and the greatest number of renewals occurred in April and October.

Figs. 4*(a)* Atlantic, and *(b)* Pacific, shows the geographical location of the individual "Liberty" vessels at the time of propeller loss, together with their approximate course, again based on Lloyd's List.

Table I is the key to Figs. 4*(a)* and *(b)* and lists the details of casualty voyages, also, where known, the loading condition at the time, with or without information as to the draughts when leaving the last port of call.

As would be expected from considerations of traffic density, the north Atlantic area includes by far the greatest number of casualties. Of the world total of 101 casualties, 55 occurred in the north Atlantic north of about latitude 30 deg. N., and of these no less than 52 were west-bound, and although information is lacking in the case of 3 vessels, at least 47 were on ballast voyages at the time. Of the "Liberty" world total casualties enquiries have revealed that no less than 81 occurred when in ballast, despite information being lacking in the case of 8 vessels.

FIG. 4(b)—*Geographical location in the Pacific area of "Liberty" ships at time of propeller loss*

It is perhaps of significance that the majority of breakdowns took place when on a course contrary to the prevailing winds, the principal casualty areas being the western Atlantic and U.S. eastern seaboard, the Channel approaches, the Azores, and mid-Pacific around Honolulu.

As regards the north Atlantic casualties the balance, or rather lack of balance, of trade, prevailing since the war, together with U.N.R.R.A. and Marshall Aid, has undoubtedly caused a predominantly easterly flow of cargoes, more particularly as carried in American bottoms. Furthermore, having regard to the relative costs of bunker oil in the United States, as compared with Europe (for example, about 33⅓ percent greater in the U.K.), the majority of these vessels would bunker in the U.S. for a return voyage to Europe, the homeward leg of which would usually be made in ballast. Thus, those vessels returning from N.W. Europe and nearing the American seaboard, or those from Italy and S. France in the vicinity of the Azores, would at the time of casualty have steamed anything from 5000 to 6000 miles, which at 11 knots and 27 tons per day represents a total consumption of up to 540 tons of oil fuel. This is, of course, quite normal, but where, as is usual from considerations of contamination, oil fuel tanks are not replenished with water ballast and additional dry ballast is not carried, it will undoubtedly be found difficult in the "Liberty" type of vessel to strike a reasonable compromise between adequate propeller immersion and sufficient draught forward to give satisfactory steering qualities and minimize pounding damage.

Detailed discussion of the ballasting problem is included in a later section.

At this stage reference must be made to the almost complete immunity from screwshaft failures enjoyed by the coal-burning "Ocean" class, only one case of

a condemned shaft having been reported, and this after a period of six years service. It may also be significant that this particular vessel had been converted to oil fuel seven months previously.

The "Ocean" class vessels were built in the U.S. to the order of the British Government over the period 1941-42. The total number built was 60, of which 18 became war casualties and the remaining 42 have been operating for an average period of about 6½ years, mostly under British ownership, whereas the bulk of the "Liberty" casualties have occurred after a very much shorter period of service, in some cases only a few months.

THE CHARACTER OF SCREWSHAFT FAILURES

Fig. 5 shows details of the "Liberty" type propeller shaft tail end, from which it will be noted that the design follows normal practice and incorporates the modern sled-runner type keyway. A standard type of rubber-ring sealing arrangement is also indicated between the end of the bronze liner and the usual recess in the propeller boss. It should be noted that precisely similar keying and sealing arrangements were used on the "Ocean" class screwshafts.

In the Canadian "Victory" vessels, however, the semi-circular ended type of keyway was usually adopted, but it should be noted these shafts were by no means immune from failure as will be seen from Table II. A study of this table indicates that a common type of failure is characterized by cracking parallel to one or both faces of the keyway, more usually on the driving side, sometimes associated with cracks radiating from the forward end of the keyway, again usually, but not invariably, on the driving side. The parallel cracks are mostly situated about ¾ inch from the side of the keyway and frequently begin circumferentially for a short distance before turning parallel to the latter. They are sometimes accompanied by axial cracking along the root fillet (if any!) of the keyway (see ships O, T, and V, Table II), and the material between the parallel crack and the keyway is in some cases honeycombed with fine cracks. Fig. 6 is a representative example of this type of failure (ship T), which, as shown in Table II, is almost invariably accompanied by a badly fitted propeller and, or alternatively key and not infrequently also by evidence of corrosive attack. In the latter case examination of the rubber sealing ring usually reveals the cause.

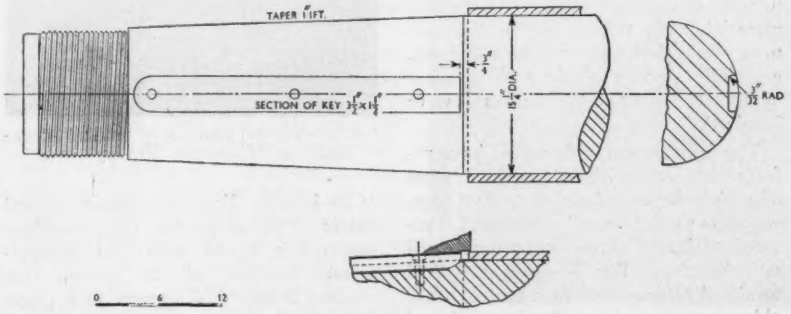


FIG. 5—*Details of "Liberty" propeller shaft tail end*

Where the propeller has been slack on the taper, it is also common to find a bronze deposit on the shaft surface, indicative of rubbing or fretting action between the surfaces. This class of failure bears strong symptoms of excessive bearing pressure on the sides of the keyway at the forward end, most probably due to heavy fluctuations of tangential loading, which in the absence of frictional grip in the slack taper fit, have to be carried by the sides of the keyway alone. Where no proper fillet is provided at the root of the keyway, this type of cracking is facilitated.

As regards the cracks radiating from the forward end of the keyway these may be inclined at from 45 deg. to 60 deg. to the shaft axis and reach several inches in length. Cases also occur where circumferential cracking takes place in way of the sealing ring, particularly where the latter has permitted sea water leakage and, or alternatively where the shaft has been nicked circumferentially on account of careless machining of the liner end. It is of interest to note that micro-examination of some of the cracks described in Table II showed them usually to be trans-crystalline in character; further, the micro-structure of the metal was fine-grained and uniform. In addition, mechanical tests showed the material to have a tensile strength of about 30 ton per sq. in. with a yield strength of 14 ton per sq. in., and a reversed bending endurance limit of $\pm 13\frac{1}{2}$ ton per sq. in. The possibility of defective material being responsible for the failures would thus appear to be ruled out, especially since a considerable number of different forges and steelworks were involved.

The failures so far described have referred to Canadian-built vessels, in most of which the round-ended type of keyway was used. Some illustration of fractured "Liberty" type screwshafts will now be given. Fig. 7 shows a typical torsional fatigue type of fracture, probably emanating from the forward end of the keyway. From the appearance and extent of the oyster-like ring markings reported it would seem that approximately one-half of the section was cracked before final rupture took place. The vessel was two days out from Ant-

FIG. 6.—Typical example of cracked screwshaft in "Canadian Victory" type.

TABLE II—Details of Typical Screwshaft Failures in Canadian "Victory" and "Canadian" Type Vessels

Ship	Condition of sealing ring	Compression of sealing ring	Fit of propeller on taper	Fit of key in keyway	Was a radius given to keyway at root?	Location of cracking: Parallel to keyway	Location of cracking: Radiating from forward end of keyway	Location of cracking: Circumferentially at end of liner	Remarks
A	Good	Slight	Good	Slack	—	Yes (D) 8½ inch	—	—	—
B	Good	Light only	Slack	Slack	No	Yes (D)	Yes (D) 45 deg.	—	Some pitting
C	Good	—	Slack	—	—	Yes (L) 4 inch	Yes (D) 60 deg.	—	Moisture oozed from cracks
D	—	—	Slack	Slack	—	Yes (D) 2½ inch	Yes (D)	—	Some pitting
E*	Bad	Slight	Poor	Poor	—	—	—	Yes	Some corrosion, shaft nicked at end of liner
F	Bad	—	—	—	—	—	—	Yes	Corrosion
G	Good	Good	Good	Good	—	—	Yes (D)	—	—
H	Good	Good	Good	Good	—	—	Yes (D)	—	Sled type keyway
I	Good	Good	Slack	Slack	—	Yes (D) 7 inch	—	—	—
J*	—	—	Good	Good	—	Yes (D) 7 inch	—	—	—
K	—	—	Good	Good	—	—	Yes (L)	—	—
L	Bad	—	—	—	—	—	Yes (D)	—	Scarphed joint in rubber ring failed
M	Good	—	Good	Good	—	—	Yes (D)	—	—
N	Good	—	Good	Good	—	—	Yes (D)	—	—
O	Bad	Light only	Slack	Slack	Poor	Yes (D) 3 inch	Yes (D) and (L)	—	Bad corrosion; bottom of keyway cracked full length (D)
P	Good	—	Good	Good	—	—	Yes (L)	—	—
Q	—	—	Slack	—	—	Yes (D) and (L) 9 inch	—	—	—
R	Good	Nil	Poor	Slack	—	Yes (D) and (L) 3½ inch	—	—	—
S	—	—	Slack	—	—	Yes (D)	Yes (D)	—	—
T	—	—	Slack	Poor	Poor	Yes (D) 8 inch	—	—	Pitted, also cracked in root fillet
U	—	Slight	—	Slack	—	—	Yes (D) and (L) 45 deg.	—	—
V*	—	—	Slack (nut slack also)	Slack 1/32 in.	—	Yes (D) 14 inch at root	—	—	After heavy weather in light condition

*="Canadian" type vessel, Scotch boilers, oil-fired, no deep tanks forward.

Notes.—(1) (D)=driving side of keyway when running ahead.
(L)=leading side of keyway when running ahead.
(2) All keyways of round-ended type, except where stated.

werp for New York in ballast with a leaving draught aft of 17 ft. 8 in. (see Table I and Fig. 4, vessel No. 66). Fig. 8 is a beautiful example of the disastrously weakening effect of corrosion under the action of repeated stress accentuated by different kinds of stress-raisers. The fatigue cracks at both corners of the sled-type keyway, also on each side of the tapped hole for the forward key-retaining bolt, will be noted, the inclination of the cracks being something like 60 deg. to the shaft axis. It should be stated that apart from the cracks shown, the root of the keyway was cracked for its full length on the driving side and part length on the leading side. No fillet radius had been given to the keyway. The forward part of the root crack on both sides, when opened up, presented a zig-zag or saw-tooth appearance, clearly indicative of torsional fatigue stress, and this was

Fig. 7.—Typical example of screwshaft failure in "Liberty" type.

confirmed by the appearance of a number of the characteristic corrosion-fatigue star cracks, again at about 60 deg. to shaft axis after hot etching the bottom of the keyway abreast the tapped hole. The shaft was also heavily corroded underneath the leaky rubber ring and was cracked circumferentially all round the shaft, except at the end of the keyway, to a depth of up to ¼ inch. Metallurgical examination of the shaft material, including mechanical tests, showed that the steel was of good quality, with a tensile strength of 30 ton per sq. in., 34 percent elongation in 2 inch and over 30 ft-lb. Izod impact value. In this example it would appear from the orientation of the fatigue cracks that combined bending and torsional stresses were responsible for the failure.

The next example, Fig. 9, shows a corrosion-fatigue crack at the end of the shaft liner. The fissure can be seen running partly circumferentially and partly inclined to the shaft axis.

Figs. 10 and 11 show different views of another "Liberty" shaft which broke at sea, and here the familiar fatigue ring markings are clearly in evidence, indicating that the point of initiation of the crack was roughly at 90 deg. from the keyway, i.e., in line with one of the propeller blades. The fatigue crack, which extended over more than half the cross section, was only slightly helical, and suggests that combined bending and torsional stresses took part in bringing about failure. This vessel lost her

FIG. 9.—Typical crack patterns in "Liberty" screwshafts.

FIG. 8.—Typical crack patterns in "Liberty" screwshafts.

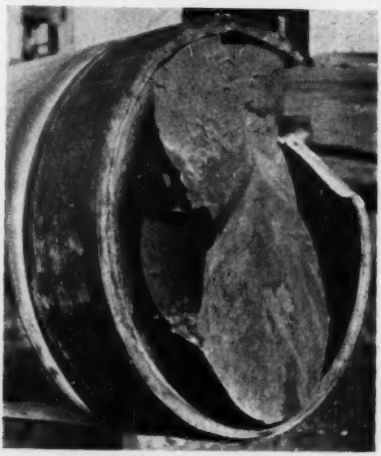


FIG. 10.—Fractured "Liberty" screwshaft (see also Fig. 11 for end view).

FIG. 11.—Fractured "Liberty" screwshaft (see also Fig. 10 for side view).

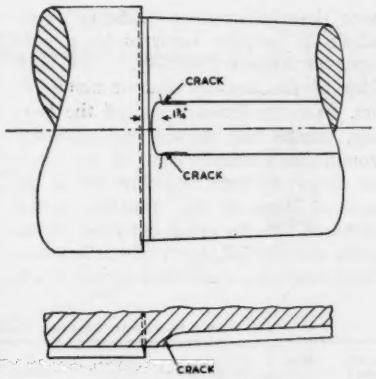


FIG. 12—*Cracked tail shaft, S.S.* Bendoran

propeller when two days out from Rouen for Hampton Roads. She was in ballast with a mean draught between 14 ft 6 in. and 16 feet (see Table I and Fig. 4, vessel No. 24).

Fig. 12 illustrates another type of "Liberty" fracture, the cracks luckily being detected at an early stage of their development and then only by the application of magna-flux methods, being invisible to the naked eye or to the glass, except when highly polished and under very strong oblique lighting. In this case, which is apparently typical of a fair number of "Liberty" failures, there was evidence that the propeller was a slack fit on the taper at the forward end and the key was also a poor fit in the shaft. It will be noted that the cracks run along the root of the keyway (very small fillet radius was provided) for about 1 inch on both sides before extending through the sides of the keyway in a circumferential direction. The fracture probably corresponds to the early stages of that shown in Fig. 6 for the round-ended keyway in the Canadian "Victory" shaft. In the case of Fig. 12 the shaft was four years old and had been drawn for examination only six months previously. There was no indication of corrosive action.

It may be of interest to quote one

TABLE III—SUMMARY OF I-NODE, THIRD ORDER STRESSES RECORDED BY TORSIOGRAPH

No.	Name of vessel	Type	Torsiographed by	3rd order critical		Condition of vessel	Draughts	
				r.p.m.	Screwshaft stress, lb. per sq. in.		Forward ft. in.	Aft ft. in.
1	S.S. *Samoa*	"Liberty"	Harland and Wolff	77	± 3,000	In ballast	13 0	15 6
2	S.S. *Benarty* ex *Ammia*	"Liberty"	Lloyd's	76	± 2,850	Part loaded	—	17 3
3	S.S. *Clan Macfadyen**	"Liberty"	Lloyd's	76·5	± 4,000	In ballast	9 0	13 6
4	S.S. *Ira Nelson Morris*†	"Liberty"	G.E.C. (for A.B.S.)	78·5	± 4,400	In ballast	8 0	14 6
5	S.S. *Clan Macbeth**	"Ocean"	Lloyd's	77	± 3,400	In ballast	8 6	13 6
6	S.S. *Bendoran*	"Liberty"	Lloyd's	76·5	± 3,000	In ballast	7 0	13 0

Note.—All stresses derived from Geiger torsiograph records except where stated.
* In these vessels Geiger stresses were confirmed by simultaneous recording from electrical resistance strain gauges (see Fig. 15).
† In this vessel stresses were recorded electrically.

more description of a "Liberty" type failure as actually reported by a surveyor in August 1948:—". . . on both sides of the keyway and at about 25 mm. from the forward end of the keyway, cracks had started and extended around the circumference of the shaft for about 45 mm., running aft at an angle of about 45 deg. Further, on the starboard side the crack extended downwards for the full depth of the keyway. There were also signs of a second crack on the starboard side at about 25 mm. aft of the main crack. . . . There were signs of the key having borne heavily on the forward starboard side of the keyway. . . ." This would seem to be a clear case of torsional fatigue loading, aggravated probably by a badly fitted key. That the cracks occurred on both sides of the keyway strongly suggests the action of an alternating tangenital stress such as could be set up during heavy torsional oscillation.

TABLE IV—NATURAL FREQUENCY TABULATION: ONE-NODE TORSIONAL VIBRATION

No. of mass	Mass	J/g, lb. inch sec.^{2}	$J/g\ p^2$, lb. in.	θ radians	$J/g\ p^2\ \theta$, in. lb.	$\Sigma J/g\ p^2\ \theta$, in. lb.	C, lb. in. per rad.	Twist, radians
1	Valve gear	245	$0{\cdot}143 \times 10^6$	1·0000	$0{\cdot}143 \times 10^6$	$0{\cdot}143 \times 10^6$	$1{,}390 \times 10^6$	0·0001
2	h.p. cylinder	14,705	$8{\cdot}5900 \times 10^6$	0·9999	$8{\cdot}590 \times 10^6$	$8{\cdot}733 \times 10^6$	$1{,}330 \times 10^6$	0·00656
3	Valve gear	245	$0{\cdot}1430 \times 10^6$	0·99334	$0{\cdot}142 \times 10^6$	$8{\cdot}875 \times 10^6$	$1{,}330 \times 10^6$	0·00667
4	m.p. cylinder	15,335	$8{\cdot}9500 \times 10^6$	0·98667	$8{\cdot}830 \times 10^6$	$17{\cdot}705 \times 10^6$	$1{,}337 \times 10^6$	0·01325
5	Valve gear	295	$0{\cdot}1722 \times 10^6$	0·97342	$0{\cdot}1678 \times 10^6$	$17{\cdot}8728 \times 10^6$	$1{,}108 \times 10^6$	0·01612
6	l.p. cylinder	18,343	$10{\cdot}7100 \times 10^6$	0·95730	$10{\cdot}2500 \times 10^6$	$28{\cdot}1228 \times 10^6$	$1{,}542 \times 10^6$	0·01821
7	Turning gear	3,557	$2{\cdot}0760 \times 10^6$	0·93909	$1{\cdot}948 \times 10^6$	$30{\cdot}0708 \times 10^6$	$23{\cdot}6 \times 10^6$	1·27300
8	Propeller	154,000	$90{\cdot}000 \times 10^6$	−0·33390	$-30{\cdot}07 \times 10^6$	0	—	—

F=230·6 v.p.m. $p^2=0{\cdot}000584 \times 10^6$.

S.S. *Bendoran*

Built-up crankshaft. Manganese-bronze propeller, partially immersed.
(Draught aft approximately 13 feet)

INVESTIGATIONS INTO THE CAUSES OF FAILURE

(A) Torsional Vibration

As soon as it was suspected that the outbreak of screwshaft failures in American war-built vessels could be attributed to systematic causes, and after careful study of the nature of the reported fractures, it was decided to investigate the torsional vibration characteristics of the shafting system. Detailed calculations were therefore made on both sides of the Atlantic and agreement was reached that 1-node natural frequency was of the order of 230 Rpm. for the most usual combination of manganese bronze propeller and fully built crankshaft (a minority were fitted with cast-iron propellers) (see Table IV). It was thus apparent that the third order critical speed could be expected at about 76 Rpm., i.e., coincident with the maximum service speed of the engines. Attempts were then made by various methods of calculation to estimate the maximum stress in the screwshaft resulting from operation on this critical at resonance. However, the results were found to vary widely, mainly on account of uncertainty as to the appropriate value of the engine damping factor. Reference to the normal elastic curve (Fig. 13*(b)* will show that in the steam reciprocating engine type of installation, the total engine moment of inertia is considerably less than that of the propeller, and therefore the relative amplitude of vibration at the engine for the 1-node mode may be as much as three times that at the propeller. In oil engine installations with machinery amidships, on the other hand, the reverse usually obtains and the propeller amplitude may be three to five times that at the engine, or even more. In the steam engine installation, therefore, the influence of engine damping becomes correspondingly more important in estimating vibration stresses, and for the same reason a more

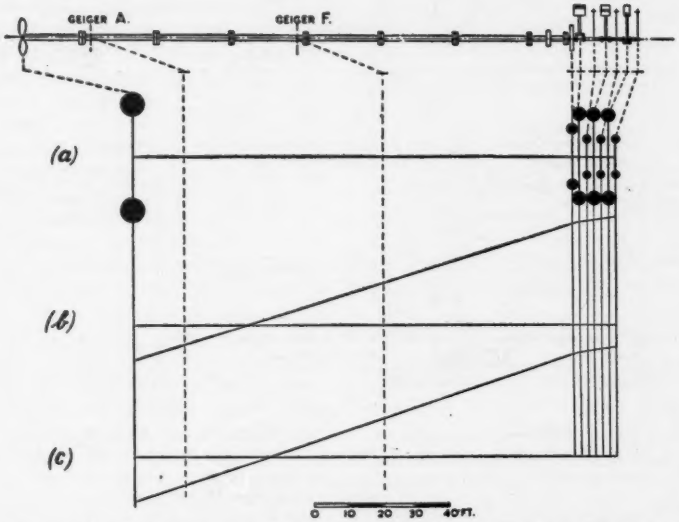


FIG. 13(a) (b) (c)—*Shafting arrangement and normal elastic curves for 1-node mode of torsional vibration, showing location of torsiographs for tests on S.S.* Bendoran

accurate knowledge of the harmonic exciting torques from the engine cylinders is desirable.

Torsiograph measurements were therefore made on a number of "Liberty" vessels on both sides of the Atlantic and the recorded results of some of these tests are summarized in Table III. It will be seen that the calculated 1-node natural frequency has been confirmed and the measured stresses range from $\pm$ 2850 lb. per sq. in. to $\pm$ 4400 lb. per sq. in. It will also be noticed that vessel No. 5, S.S. *Clan Macbeth*, is of the "Ocean" type which, as previously mentioned, have identical engine-shafting-propeller systems to the "Libertys." The measured stress of $\pm$ 3400 lb. per sq. in. in this vessel compares with the average of $\pm$ 3450 lb. per sq. in. for the five "Libertys," thus confirming the similarity of the two types as regards torsional vibration stresses.

In the case of vessel No. 1 difficulty was experienced in forcing the engines above 77 Rpm., and as a result some doubt remained as to whether the peak stress had in fact been reached.

This uncertainty was dispelled in vessel No. 2, when a maximum engine speed of 80 Rpm. was attained, at which the measured stress had fallen to $\pm$ 2600 lb. per sq. in. In both these vessels the records were taken by means of a single Geiger torsiograph, belt driven from the intermediate shafting at several points. With this method of non-synchronous recording no account is taken of the fact that as a result of propeller damping there will, in general, be a phase difference between the records taken at any two positions along the shafting which is not exactly 0 deg. or 180 deg. (depending upon whether the two positions are on the same or opposite sides of the theoretical nodal position). Such a phase difference persists at resonance and therefore to derive the true amplitude of twist in the shafting it is necessary to use the measured vector difference of the two amplitudes. (See Appendix I.)

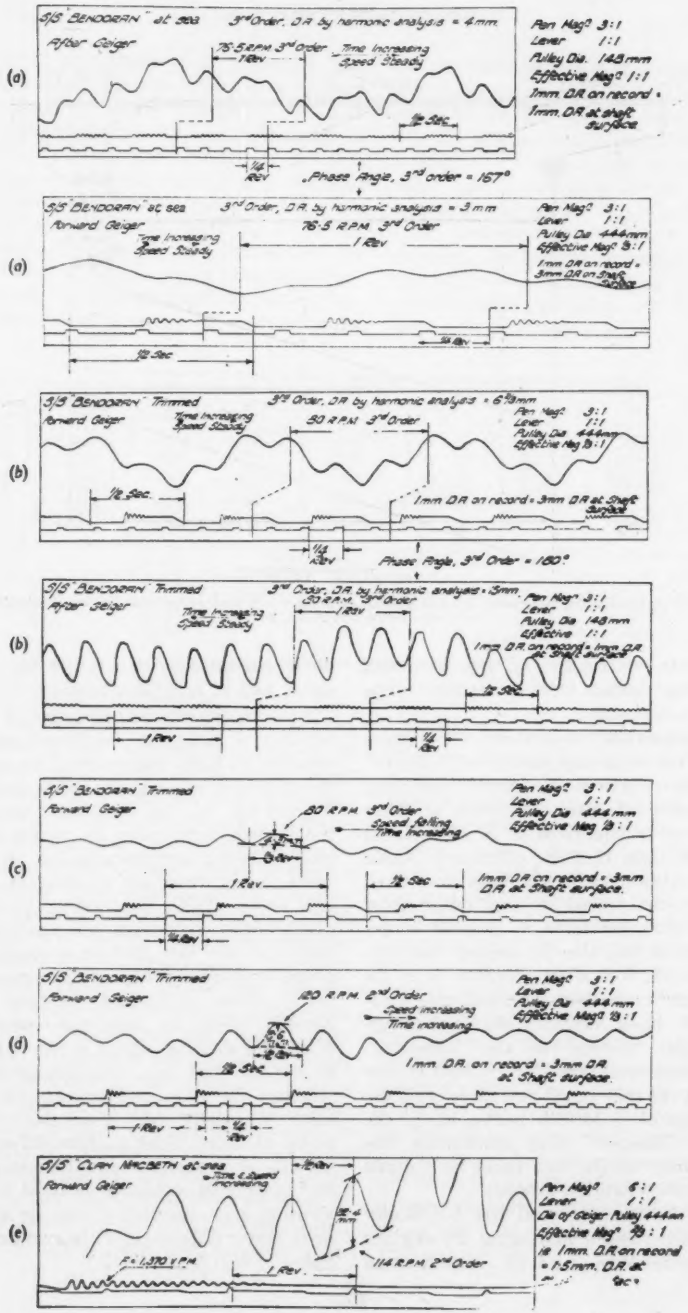


FIG. 14—*Geiger torsiograph records S.S. Bendoran and S.S. Clan Macbeth*

With these considerations in mind, therefore, the stresses recorded by Lloyd's Register in the case of the remaining three vessels in Table III, i.e., Nos. 3, 5, and 6, were all derived from amplitudes measured simultaneously by two phased Geiger instruments driven from two different points on the line shafting (see, for example, Figs. 13 and 14 for vessel No. 6). Further, as a check on the accuracy of this procedure, in the case of vessels Nos. 3 and 5 simultaneous electrical recordings were made by means of an a.c. bridge network of electrical resistance strain gauges secured to the shaft surface in the usual 45 deg. arrangement, the connections being taken off through special slip rings and brush gear. The output from the bridge, after suitable amplification, was fed into an M.I.T.,* photographically recording oscillograph. Fig. 15 shows a typical example of M.I.T. film record for the 3rd order critical speed on the S.S. *Clan Macfadyen*.

For electrical recording of the relatively low frequencies of torsional oscillation characteristic of marine shafting installations, the a.c. bridge technique is strongly to be recommended if trouble-free operation and accurate results are required. The drawback to the ordinary d.c. excited bridge method is, of course, the difficulty of designing stable amplifiers for such low frequencies.

As a proof of the accuracy of both methods of recording employed, it can be stated that for vessels Nos. 3 and 5 the results showed agreement in stress

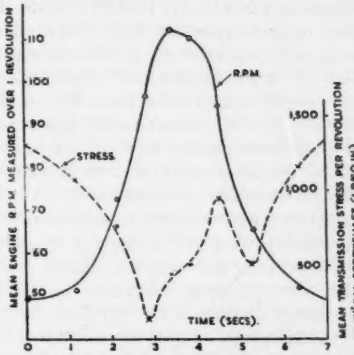


FIG. 16—*Variations of engine speed and mean transmission stress in screwshaft, S.S.* Clan Macbeth *during racing*

within 4 percent and 2 percent respectively, between the two methods, 24-ordinate harmonic analysis being employed both for M.I.T. and Geiger records. The higher figure is quoted in Table III in each case.

During the trials on the S.S. *Clan Macbeth* on passage from London to Glasgow a south-westerly gale was encountered in the Channel and with the vessel drawing only 13 ft., 6 in. aft, severe racing occurred. The opportunity was therefore taken to record the torsional stresses in the shafting and the engine speed variations, using both the Geiger torsiograph and the M.I.T. electrical equipment. Fig. 16 shows curves of mean engine speed per revolution and mean transmission stress in screwshaft per revolution, as derived from strain gauge and timing traces. It is

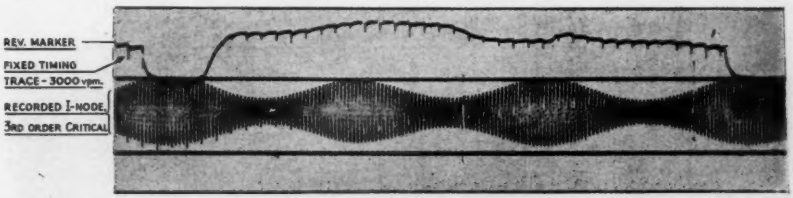


FIG. 15—*M.I.T. film, S.S.* Clan Macfadyen

* M.I.T. = Massachusetts Institute of Technology.

significant to observe that even with the Hp. engine expansion link shut well in and with an efficient governor set to cut off steam at about 85 Rpm., speed surges still occurred between 50 and 112 Rpm. This is a fundamental limitation of non-anticipating type of governor, in of the torsional vibration strains in the intermediate shafting so far as the expansive power of the steam remaining in the Hp. receiver is always available to accelerate the engines. Fortunately, the corresponding recorded variations of mean stress in the screwshaft per revolution only ranged from 150 to 1300 lb. per sq. in. While recording with the forward Geiger torsiograph, however, the flank of the 2nd order, 1-node torsional vibration critical was picked up between about 100 and 114 Rpm., at which latter speed it was approaching resonance (see Fig. 14(E)). The corresponding vibration stress in the screwshaft was calculated at ± 8200 lb. per sq. in. (see Appendix II). Previous calculations had indicated that the resonant speed for this critical, assuming the screw to be completely out of water (and thus without the usual 25 percent increase in moment of inertia to allow for entrained water), would be between 118 and 120 Rpm. (see Table V and Fig. 13*(c)*). Further, it had been estimated that the 2nd order vibration stress at resonance, assuming no propeller damping, would be of the order of $\pm 15{,}000$ lb. per sq. in. in the screwshaft. The measured stress value of ± 8200 lb. per sq. in. at 114 Rpm. thus appeared reasonable. Unfortunately, however, no speeds higher than 114 Rpm. were recorded during the trial, and in consequence uncertainty remained as to the maximum value of the 2nd order resonant stress.

Various alternative means were considered whereby this information could be obtained, and through the courtesy of the directors of Messrs. Wm. Thompson and Co., managers of Ben Line Steamers, Ltd., Leith, it was finally decided to carry out special trials on another "Liberty" ship, the S.S. *Bendoran*. The proposed procedure was to run the engine light with the vessel trimmed by the head sufficiently to bring the screw clear of the water.

The trial party joined the vessel in Hull and took torsiograph records on ballast passage to Leith, using two phased Geiger instruments as previously. In addition a tachometer had been fitted to the main engine and in order to derive engine speeds more accurately, a special 4-per-revolution contact had been arranged on the line shafting. An accurate timing trace was also provided by means of a ½-second clock. The distance between the two torsiographs was about 60 feet and Fig. 13*(a)* shows their location on the line shafting. The draughts of the vessel during the voyage to Leith were approximately 7 feet forward and 13 feet aft. It had been hoped that heavy weather records might have been obtained on passage, but unfortunately there was fog with a flat calm sea. It was therefore necessary to fall back on the original proposal to trim the vessel on arrival at Leith.

TABLE V—NATURAL FREQUENCY TABULATION: ONE-NODE TORSIONAL VIBRATION

No. of mass	Mass	J/g, lb. in. sec.^{2}	$J/g\ p^2$, lb. in.	θ radians	$J/g\ p^2\ \theta$, in. lb.	$\Sigma J/g\ p^2\ \theta$, in. lb.	C, lb. in. per rad.	Twist, radians
1	Valve gear	245	$0{\cdot}1524\times10^6$	1·0000	$0{\cdot}1524\times10^6$	$0{\cdot}1524\times10^6$	$1{,}390\times10^6$	0·000109
2	h.p. cylinder	14,705	$9{\cdot}1400\times10^6$	0·99939	$9{\cdot}14000\times10^6$	$9{\cdot}2924\times10^6$	$1{,}330\times10^6$	0·006990
3	Valve gear	245	$0{\cdot}1524\times10^6$	0·99290	$0{\cdot}1513\times10^6$	$9{\cdot}4437\times10^6$	$1{,}330\times10^6$	0·007110
4	m.p. cylinder	15,335	$9{\cdot}5200\times10^6$	0·98580	$9{\cdot}3800\times10^6$	$18{\cdot}8237\times10^6$	$1{,}337\times10^6$	0·014100
5	Valve gear	295	$0{\cdot}1835\times10^6$	0·97170	$0{\cdot}1782\times10^6$	$19{\cdot}0019\times10^6$	$1{,}108\times10^6$	0·017150
6	l.p. cylinder	18,343	$11{\cdot}400\times10^6$	0·95455	$10{\cdot}8700\times10^6$	$29{\cdot}8719\times10^6$	$1{,}542\times10^6$	0·019380
7	Turning wheel	3,557	$2{\cdot}210\times10^6$	0·93517	$2{\cdot}0650\times10^6$	$31{\cdot}9369\times10^6$	$23{\cdot}6\times10^6$	1·35220
8	Propeller	123,200	$76{\cdot}600\times10^6$	−0·41700	$-31{\cdot}937\times10^6$	0	—	—

F=238·3 v.p.m. $p^2=0{\cdot}000622\times10^6$.

S.S. *Bendoran*.

Propeller inertia excluding entrained water.

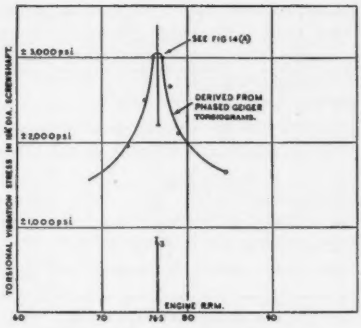


FIG. 17—*Resonance curve; 3rd order, 1-node, S.S.* Bendoran *at sea*
(Vessel's draughts: Forward 7ft., aft 13ft. approx.)

However, the 3rd order critical speed was recorded at sea, giving a stress of ± 3000 lb. per sq. in. in the screwshaft. Fig. 14(A) shows sections of the records at the resonant speed of 76-77 Rpm. and the calculations are given in Appendix II. The 3rd order resonance curve is plotted in Fig. 17.

Preliminary calculations had indicated that with the vessel fully discharged a sufficient trim by the head could be obtained by emptying the after peak tank and filling the fore peak tank, the forward double bottom tank, and Nos. 1 and 2 deep tanks, but in addition it would be necessary to flood No. 1 hold with about 600 tons.

Before proceeding with the test, careful consideration was given to the strength of the after bulkhead in No. 1 hold, also to the bending stresses in the hull girder, and to the stability of the vessel in the proposed heavily trimmed condition, bearing in mind the large free surface in No. 1 hold, and the difficulty of ensuring that all other tanks were either completely empty or pressed up. As a final check on the calculations an inclining experiment was carried out during the night before the tests with the vessel moored in Leith docks, the inclining moment being obtained by pumping 15 tons of oil fuel from one engine-room settling tank to the other. The results indicated that the stability in the trimmed condition would be adequate. Accordingly, the trimming operation was proceeded with, the final draughts being 18 ft., 5 in. forward and 3 ft. aft. Under these conditions the propeller blade tips were immersed 1 ft., 9 in., which it was judged would provide negligible damping to the torsional vibrations.

Fig. 18 shows the vessel in her trimmed condition, and Fig. 19 gives a close-up view of the vessel's stern.

The purpose of the test was simply to measure the value of the 1-node, 2nd order stress in the screwshaft at resonance under comparable conditions to those obtaining during racing at sea.

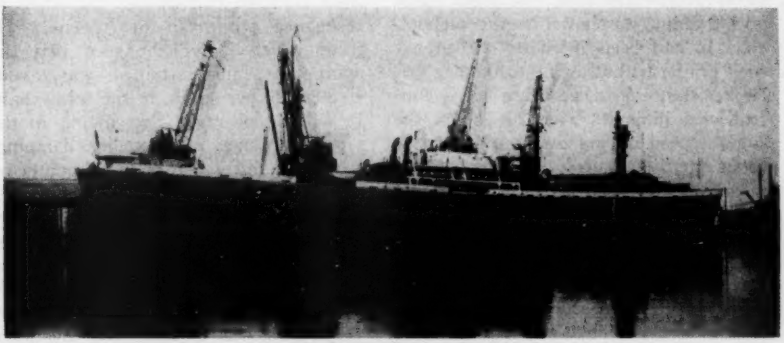


FIG. 18.—S S *Bendoran* in trimmed condition.

FIG. 19—S S *Bendoran* in trimmed condition; close-up of stern.

Accordingly, the Hp. engine expansion link was shut in to a minimum and the engine controlled on the hand throttle alone. Despite the short duration of the test, it was considered prudent to open the water service to the stern tubes. The operation of the engine controls during the race was entrusted to one of the Ben Line's senior chief engineers, in whose capable hands the engines were safely accelerated to a maximum speed of 140 Rpm., as shown by the tachometer. It had been intended not to exceed 125 Rpm., but owing to the negligible resistance load, accurate speed control was difficult. Fig. 20 shows the engine speed during the race plotted as mean Rpm. per revolution against time in seconds as derived from the timing traces on the torsiograph records. On the diagram is drawn to the same scale, for direct comparison, a portion of the corresponding graph, Fig. 16, obtained from S.S. *Clan Macbeth* during racing at sea, from which it will be seen that the accelerations in the vicinity of the

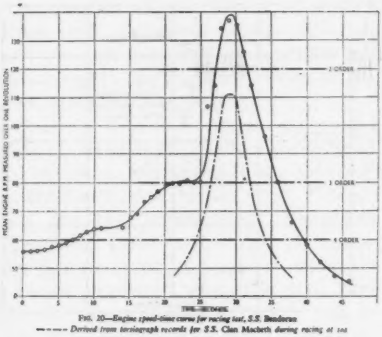


FIG. 20—Engine speed-time curve for racing test, S.S. Bendoran
——— Derived from torsiograph records for S.S. Clan Macbeth during racing at sea

2nd order critical, from say 100 Rpm. upwards, are of the same order in the two cases, the speed rise in the S.S. *Bendoran* being, if anything, somewhat slower.

During the run-up, four 1-node critical speeds were clearly picked up, i.e., the 6th, 4th, 3rd, and 2nd orders, from with the 1-node natural frequency with the propeller substantially free was measured to be approximately 240 Rpm. It will be noted from Fig. 20 that at 80 Rpm., corresponding to the 3rd order critical speed, a pause was made for about four seconds during which the vibration built up to a steady value, but that the remaining criticals were passed through fairly rapidly. Fig. 14(B) shows phased sections of the records at the 3rd order resonant speed of 80 Rpm. and the calculations for stress at resonance are given in Appendix II, from which it will be seen that the maximum value attained was about $\pm$ 8000 lb. per sq. in. in the screwshaft. It will be observed that owing to the virtual absence of propeller damping, the phase difference between points on opposite sides of the node is almost 180 deg., and the screwshaft stress is more than doubled, as compared with the measured value of this critical at sea, although the mean indicated pressure in the engine cylinders must have been very little more than the frictional mean pressure, or say 10-12 percent of

that at full power. Clearly the major source of energy input must have been the 3rd order component of the inertia torques due to the reciprocating masses (see Appendix III). The same critical speed was also picked up on the run-down, but owing to the much greater rapidity with which it was passed through (see Fig. 20), the recorded stress amounted to only ± 4900 lb. per sq. in. as determined from a record taken on Geiger F alone (see Fig. 14(C)).

Fig. 14(D) is a section of the forward Geiger record showing the passage through the 2nd order critical during the run-up. Unfortunately, the after Geiger was hitting the stops during this period, but since the phase difference was 180 deg., the stress can be readily calculated from the simple formula:—

$$f = \pm \frac{C_F \theta_F}{Z\left(1 - \frac{C_F}{J_2 p_c^2}\right)} \quad \ldots\ldots\ldots\ldots (1)$$

where f = vibration stress at resonance in 15¼ inch diameter screwshaft ($\pm$ lb. per sq. in.).

Z = sectional modulus of shaft in torsion (inch³).

θ_F = recorded amplitude of swing at forward Geiger ($\pm$ radians).

C_F = calculated torsional stiffness of shafting between propeller and Geiger F (lb.-in. per radian twist).

J_2 = mass moment of inertia of dry propeller (lb.-in.-sec.²).

p_c = phase velocity of vibration at resonance (radians per sec.).

Inserting the numerical values and the appropriate maximum value of θ_F from the record (see Appendix II), the maximum value of the 2nd order stress is found to be ± 9850 lb. per sq. in. in the 15¼-inch diameter screwshaft. In this case, owing to the acceleration of the engines and propeller, there was a mean torque at the critical of about 20 percent of full load torque.

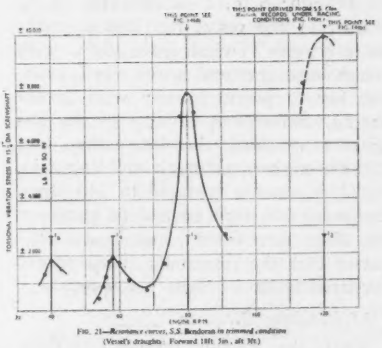


Fig. 21—Resonance curves, S.S. Bendoran in trimmed condition (Vessel's draughts: Forward 18ft. 3in., aft 20ft.)

Resonance curves for the various critical speeds are plotted in Fig. 21, but for the 2nd order only the peak value is shown with, for comparison, the stress value of ± 8200 lb. per sq. in. at 114 Rpm. as measured on the S.S. *Clan Macbeth* during racing at sea. It will be seen that these two points plot fairly well together, particularly bearing in mind that engine damping can vary widely in otherwise identical installations.

From the results of this test it is estimated that under racing conditions at sea the maximum value of the 2nd order vibration stress in the screwshaft may, on the average, be of the order of $\pm 12{,}500$ lb. per sq. in. (see Appendix III). Should, however, the peak of the speed surge happen to fall at about 120 Rpm., it is possible that even higher stresses may be built up.

To complete the story of the S.S. *Bendoran* it should be added that on changing the screwshaft in dry-dock after the tests, the old shaft was found cracked under the magnetic test, as already described in relation to Fig. 12. Further, as luck would have it, when leaving Leith again in light ballast condition (draughts leaving: 6 ft., 9 in. forward, 14 ft., 7 in. aft; arriving Mid-

dlesbrough: 8 ft., 2 in. forward, 14 ft., 5 in. aft) the vessel ran into a south-easterly gale (wind force 8-9) with rough head seas and heavy swell, causing her to pound heavily with severe racing. Subsequent inquiry of the engineers revealed the interesting fact that the engines, although well linked-in, raced repeatedly from 40 to 140 Rpm. under hand-throttle control (a governor has since been fitted), and it was estimated that the maximum speed surges occurred about 10 times per watch.

(B) Bending Stresses

That the known torsional fatigue stresses acting on the screwshafts of "Liberty" vessels are associated to some extent with bending fatigue stress is undeniable, but as to their numerical value, it is only possible to make somewhat speculative calculations at the present time as defined by more or less arbitrary assumptions. It is possible, however, that the obvious mechanical difficulties involved in their measurement by means of strain gauges will soon be overcome, in which case our knowledge of the actual stresses to which screwshafts are exposed in heavy weather will be greatly clarified.

Two calculations have been made to determine the possible order of magnitude of bending fatigue stresses in the "Liberty" screwshaft, and details are indicated in Appendix IV *(a)* and *(b)*.

Calculation *(a)* relates to the ballast condition in smooth water with a draught aft of, say, 14 ft., 7 in. and a mean draught of 11 ft., 5 in. (Condition VII, Table VI). In these circumstances, with two blades vertical the upper one will have about ⅝ of its length unloaded, and it is assumed that the reduction in unbalance caused by the inner ⅜ is negligible. The calculated stress value of ± 2540 lb. per sq. in., when added to the bending stress of ± 1600 lb. per sq. in. due to dead weight, gives a total variation of bending fatigue stress, at each of the four 90 deg. angular positions approximately defined by the blade center lines (see Fig. 11), of ± 4140 lb. per sq. in.

Calculation *(b)* is admittedly more arbitrary and is intended to explore the conditions existing when the ballasted vessel is racing heavily to speeds of upwards of 120 Rpm. in head seas, which have resulted in the vessel's speed being reduced to, say, 6 knots, probably quite a common condition in the North Atlantic in winter-time. The assumption is made that during re-immersion after a speed surge the revolutions have fallen to 100 Rpm. by the time the lower blade in its vertical position has its center of thrust at 80 percent of tip radius from the shaft axis, also that the effective thrust under these conditions is about one-half of that which would be exerted by each blade if the screw were normally submerged at the same slip. It will be noted that the resulting stress value obtained is about ± 8000 lb. per sq. in. It is realized, of course, that in practice the actual thrust would probably be reduced by cavitation effects and by the high degree of aeration of the surface water in the wake under the turbulent conditions envisaged. On the other hand, this reduction may be balanced to some extent by the impactive or blow effect as the blade enters the water at high velocity. However, the object is merely to attempt an approximation under certain specific conditions as an aid to bridging the gap between ignorance and fact.

In both calculations *(a)* and *(b)* use has been made of Troost's results for his series B.4.40, although his boss-diameter ratio was 16 percent greater giving an 8 percent smaller D.A.R. Also his blade thickness fraction of 0.045 compares with 0.0405 for the "Liberty" design. In other respects fair similarity exists.

(C) The Strength of the Keyed Propellor Assembly in Fatigue

In an attempt to assess the strength of the round-ended and sled-runner types of keyway under fatigue loading,

it may be convenient at this point to summarize briefly the results of published experimental work.

Peterson* carried out rotating-beam endurance tests on a 36 ton per sq. in. tensile normalized medium carbon steel having an unnotched bending fatigue strength of $\pm 16\frac{1}{2}$ ton per sq. in., and his results show nominal stress concentration factors of 1.61 and 1.32 for the round-ended and sled-runner type keyways respectively, i.e., a 22 percent greater reversed bending fatigue strength in the case of the sled-runner type. The diameter of the specimens, however, was only 1 inch and the keyway section $\frac{1}{4} \times \frac{1}{8}$ inch. Thus, the results could be appreciably affected by very small machining differences; further, it does not follow that the same ratio would apply in the 15¼-inch diameter shaft. Again, the tests were carried out in reversed bending only and it is by no means certain that a similar ratio would be obtained under reversed torsional stresses, or, as is probably more nearly correct for a ship's propeller shaft, under conditions of steady torsion with superimposed combined bending and torsional fatigue loading. Finally, Peterson's results did not include the effect of clamping stresses such as must be produced at the top of the taper by the edge of the propeller boss recess and by the end of the shaft liner.

As regards the latter Thum‡ has presented some comparative data from similar fatigue tests carried out in reversed bending on a soft steel of 27 ton per sq. in. tensile having an unnotched fatigue strength of $\pm 15\frac{1}{2}$ ton per sq. in. His results agree fairly well with Peterson's for the unclamped condition, i.e., stress concentration factors of 1.7 and 1.25 for the round-ended and sled-runner types respectively, or 36½ percent higher endurance limit for the latter type. However, when clamping was applied, the plain polished fatigue limit without keyway fell to $\pm 8\frac{1}{4}$ ton per sq. in. (53 percent of unclamped), and the stress concentration factors for the round-ended and sled-runner types amounted to 2.33 and 2.59 respectively, i.e., the round-ended keyway was slightly stronger. However, Thum's work is probably subject to much the same sort of limitations as Peterson's when attempting to apply the results to the full-size propeller shaft, and until large-scale tests prove otherwise, it seems there is little to choose between the two types for ship's propeller shafts. The influence of clamping stress on the other hand, whether from the end of the shrunk-on liner or from the propeller boss recess, merits closer attention and it would seem prudent to endeavor, so far as possible, to avoid the double stress concentration effect of the keyway end and the clamping stresses. It is therefore suggested that the minimum distance between the end of the keyway and the end of the liner might be limited to say, ⅛ of the screwshaft diameter. For the "Liberty" design this would represent only 3 percent increase in nominal shaft stress at the end of the keyway, allowing for the 1:12 taper.

It will have been noted from the various failures already described that in a number of cases the fatigue cracks, originating from the end of the keyway, commenced to propagate at first in a circumferential direction, suggesting at first sight the result of bending stresses. In this connection, however, it may be well to quote some work carried out by Gough† in reversed torsion on 0.4-inch diameter "Armco" iron shafts having round-ended keyways 0.109-inch and 0.055-inch wide by 0.036-inch deep. He reported that for unclamped specimens in every case fracture commenced by a crack occurring at, and developing

* Peterson, R. E. 1932. Proc. Am. Soc. Test. Mat., Vol. 32, part 2, p. 413, "Fatigue of Shafts having Keyways.

‡ Thum, A. 1935. Trans.N.E.C.Inst. Eng. and Shipbuilders, Vol. 51, p. D.122 (discussion)

† Gough, H. J. 1924. Brit. Aero. Research Comm. Reports, Vol. 2, p. 488.

along, a junction of the bottom and one side of the keyway (root fillets 0.001 inch to 0.005 inch). When the section was reached at which the shaft diameter increased (i.e., where the keyway end met the fillet radius from the grip portion), the crack usually developed in a circumferential direction. Again, when the specimens were clamped with a keyed-on boss, the cracks started just inside the boss axially along both junctions of the bottom and sides of the keyway and finished up by a circumferential type of failure. The stress concentration factors for keway alone and keyway clamped amounted to 1.14 and 1.5 respectively, i.e., about 24 percent additional reduction in fatigue strength due to clamping. Similar experiments carried out on a 0.65 percent carbon steel, without clamping and with or without keyways, gave fractures which in all cases were helicoidal, sometimes double helicoidal.

The above results tend to show that in a relatively soft mild steel, such as a propeller shaft forging, the presence of a circumferential crack does not necessarily imply that bending stresses are always responsible, although it is reasonably certain that where approximately 45 deg. helicoidal fractures occur, torsional stresses have played a major part.

Recent experiments by Dorey‡ on the strength of mild steel marine shafting in reversed torsion have indicated that the presence of a round-ended keyway, without clamping, in a 10½-inch-diameter shaft may reduce the nominal fatigue strength to about ± 10,000 lb. per sq. in. Assuming that the effect of the clamping stresses is to reduce this latter figure by a further 25 percent, it is estimated that the nominal fatigue strength of a well-fitted propeller shaft keyed assembly may be of the order of ± 7500 lb. per sq. in. for reversed torsional stresses acting alone. It has been shown by Gough and Pollard‖ that when reversed bending stresses act upon plain polished small-scale mild steel specimens subjected simultaneously to reversed torsional stresses, the value of the applied torsional shear stress at the fatigue limit for the combined conditions is reduced below that which could be withstood under reversed torsional shear stress acting alone, a similar effect being found for bending fatigue strength. In these circumstances, it would seem prudent to reduce both bending and torsional fatigue loading on propeller shafts to the minimum, especially where these may act simultaneously.

Under steady conditions in smooth water, with some bending fatigue present, such as when the propeller is partly immersed in the ballasted condition, it is estimated from the evidence so far available that the fatigue strength of a well-fitted propeller assembly, expressed in terms of nominal reversed torsional stress, may be of the order of ± 6000 lb. per sq. in. It should always be borne in mind, however, that this estimated value may be seriously reduced in service on account of a variety of causes, such as extremes of stress concentration arising from machining errors, badly fitted parts, corrosive environment, and previous overstress. The latter is particularly important and will now be considered.

(D) The General Effect of Overstress on Fatigue Strength

Whatever may be the actual magnitude of the bending fatigue stresses it seems clear that under conditions of severe racing appreciable torsional and bending fatigue loading will be applied to the screwshaft.

It is a well-established fact that although mild over-stressing in fatigue may actually raise the endurance limit

‡ Dorey, S. F. 1948. Proc. I.Mech.E., Vol. 159, p. 399, "Large Scale Torsional Fatigue Testing of Marine Shafting."

‖ Gough, H. J. and Pollard H. V. 1935. Proc. I.Mech.E., Vol. 131, p. 3, "Strength of Metals under Combined Alternating Stresses."

of mild steel, the reverse applies when the combination of amount and numbers of cycles of previous overstress exceeds certain limiting values. For most materials it is possible to plot on the usual S-N diagram what is known as a "damage line," which expresses the limiting number of cycles of a given overstress which the material can withstand without failure occurring in less than 10^7 cycles when re-tested at the virgin fatigue limit. For many materials this damage line has a form similar to the S-N curve, but with the high stress end pulled down. Some cases have been reported where the damage line almost coincides with the S-N curve, indicating large capacity for absorbing punishment by overstress without damage. In other cases the damage line may be substantially horizontal from which it may be concluded that even the smallest amount of overstress applied for a very few cycles will produce damage. A good example of the latter type of behavior has been reported by Russell and Welcker § for a 0.35 percent annealed carbon steel having a reversed bending fatigue strength of $\pm$ 16 ton per sq. in., although it is stated that some improvement was obtained by suitable heat treatment.

A particularly important series of tests on the effect of overstress in reducing fatigue strength has been carried out by Müller-Stock ¶ and others. The material investigated was a low carbon steel (St. 37) having a reversed bending fatigue strength of $\pm$ 13.8 ton per sq. in. The material withstood 30,000 cycles of stress at an overstress of $\pm$ 20.3 ton per sq. in. It was found that when different batches of test pieces were all previously overstressed by 45 percent at $\pm$ 20 ton per sq. in., but each batch for a different number of cycles, say, 5000, 10,000, etc., then when re-tested to determine the respective batch fatigue limits, the new S-N curves were all substantially parallel to the virgin curve, but increasingly below it depending upon the number of cycles of previous overstress. Thus, at 25,000 cycles of overstress at $\pm$ 20 ton per sq. in. the fatigue limit had fallen as low as $\pm$ 10 ton per sq. in., or only 72 percent of the virgin fatigue strength.

It therefore seems clear that where a shaft may be subjected periodically to cycles of comparatively heavy overstress, whether torsional, bending, or combined, the ability of the material to withstand continuously otherwise quite moderate fatigue stresses may well be seriously impaired.

(E) The Ballasting Problem in the "Liberty" Ship

From an examination of Fig. 4 and Table I there is strong evidence to suggest that the high percentage of screwshaft failure in the "Liberty," Canadian "Victory," and similar oil-burning types may be closely linked with the difficulty of ensuring adequate propeller immersion when in ballast.

Bearing in mind the almost complete immunity of the coal-burning "Ocean" vessels, it was decided to compare the two types by calculating the draughts under a number of different ballast conditions. The results of these calculations are summarized in Table VI and the following notes refer to the various conditions assumed therein:—

"LIBERTY" Type

Condition I.

Full bunkers including No. 3 deep tank.

Forward deep tanks empty.

Condition II.

As for Condition I, but showing effect of adding 420 tons water ballast in No. 2 deep tank forward.

§ Russell, H. W. and Welcker, W. A. 1936. Proc. Am. Soc. Test. Materials, Vol. 36, part 2, p. 118.

¶ Müller-Stock, H., Gerold E. and Schultz, E. H. 1938. Archiv. F. Eisenhüttenwesen., Vol. 12, p. 141.

TABLE VI

Condition	"Liberty"									"Ocean"			Condition
	I	II	III	IV	V	VI	VII	VIII	IX	I	II	III	
Light weight (tons)	3,380	3,380	3,380	3,380	3,380	3,380	3,380	3,670	3,670	3,248	3,248	3,248	Light weight (tons)
Crew and stores	30	30	30	30	24	24	24	30	30	32	32	26	Stores
Fresh water	56	56	56	56	10	10	10	56	56	54	54	10	Fresh water (No. 4 Tw. Dk.)
No. 1 D.B. tank (O.F.)	132	132	132	132	—	—	—	144 (S.W.)	144 (S.W.)	9	9	—	Fresh water (No. 5 Tw. Dk.)
No. 2 D.B. tank (O.F.)	313	313	—	313	—	—	—	340 (S.W.)	340 (S.W.)	136	136	136	No. 1 D.B. tank (S.W.)
No. 3 D.B. tank (O.F.)	232	232	—	232	—	—	—	234	—	332	332	332	No. 2 D.B. tank (S.W.)
No. 4 D.B. tank (R.F.W.)	132	132	132	132	66	66	66	132	132	158	158	158	No. 3 D.B. tank (S.W.)
No. 5 D.B. tank (O.F.)	232	232	232	232	—	—	—	236	—	109	109	109	No. 4 D.B. tank (S.W.)
No. 6 D.B. tank (O.F.)	108	108	108	108	108	108	108	110	—	114	114	48	No. 6 D.B. tank (R.F.W.)
No. 3 deep tank (O.F.)	760	700	700	160	—	760 (S.W.)	760 (S.W.)	760 (S.W.)	760 (S.W.)	246	246	246	No. 7 D.B. tank (S.W.)
E.R. settling tanks (O.F.)	102	102	102	102	102	102	102	100	100	115	115	115	No. 8 D.B. tank (S.W.)
Fore peak tank (S.W.)	145	145	145	145	145	145	145	145	145	734	—	734	Deep tank abaft E.R. (S.W.)
After peak tank (S.W.)	155	155	150	155	155	155	155	155	155	124	124	124	Fore peak tank (S.W.)
No. 1 deep tank (S.W.)	—	—	—	—	—	—	—	—	—	166	166	166	After peak tank (S.W.)
No. 2 deep tank (S.W.)	—	420	—	—	—	—	420	420	420	955	955	—	No. 3 lower hold (bunkers)
No. 2 hold (dry ballast)	—	—	—	—	—	—	—	360	360	602	602	—	Second deck bunker (coal)
No. 3 hold (dry ballast)	—	—	—	—	—	—	—	615	615	500	500	280	Cross-bunker (coal)
No. 4 hold (dry ballast)	—	—	—	—	—	—	—	525	525	—	—	—	—
Ballast displacement (tons)	5,717	6,137	5,172	5,177	3,990	4,730	5,170	8,032	7,452	7,634	6,900	5,732	Ballast displacement (tons)
Mean draught in S.W.	ft. in. 12 5	ft. in. 13 5	ft. in. 11 5	ft. in. 11 5	ft. in. 9 1	ft. in. 10 7	ft. in. 11 5	ft. in. 16 10	ft. in. 15 8	ft. in. 16 0	ft. in. 14 8	ft. in. 12 6	Mean draught in S.W.
Trim by stern	8 10½	4 04	11 5½	5 10	6 11	11 3½	6 4	8 10½	6 4½	7 9	4 1½	8 8	Trim by stern
Draught forward	7 11½	11 5	5 8	8 6	5 7½	4 11	8 3	12 4½	12 6	12 2	12 7½	8 1½	Draught forward
Draught aft	16 10	15 5½	17 1½	14 4	12 0½	16 2½	14 7	21 3	18 10½	19 11	16 8½	16 9½	Draught aft

Condition III (end of 6000-mile round voyage).

As for Condition I, but 545 tons oil fuel burnt from double-bottom tanks Nos. 2 and 3 and not replaced by water ballast.

Condition IV (end of 6000-mile round voyage).

As for Condition I, but 540 tons oil fuel burnt from No. 3 deep tank.

Condition V.

Represents extreme condition at end of very long voyage (say after 60 days steaming at 11 knots). Only No. 6 double bottom and engine-room settling tanks left (210 tons = say 8 days steaming).

One-half of reserve feed water used and reduced drinking water and stores.

No water ballast in any oil fuel tanks.

Condition VI.

As for Condition V, but showing effect of filling No. 3 deep tank with water ballast to improve propeller immersion.

Condition VII.

As for Condition VI, but showing effect of filling No. 2 deep tank with water ballast to improve immersion forward.

Condition VIII.

Special scheme of ballasting introduced by U.S.M.C. during war for North Atlantic convoys. Includes 1500 tons dry ballast in No. 2, 3, and 4 holds and 680 tons oil fuel, mainly in double bottom.

Condition IX (end of 6000-mile round voyage).

As for Condition VIII, but with double bottom fuel tanks Nos. 3, 5, and 6 burnt out, leaving 100 tons in settling tanks only (about 4 days steaming).

"OCEAN" Type

Condition I.

Full bunkers (2057 tons), including No. 3 lower hold.

Full water ballast, including 734 tons in deep tank abaft engine-room.

Condition II.

As for Condition I, but without deep tank, in order to reduce compressive stress in deck plating in sagging condition.

Condition III (end of long voyage, say 44 days).

Fuel almost burnt out, only 300 tons left in cross-bunker (7½ days steaming).

All water ballast in, including deep

tank; otherwise corresponds to "Liberty," Condition VI.

For reasonable immersion forward, however, the comparison should be with "Liberty," Condition VII.

For the "Liberty" type, comparing Conditions I and II, it is seen that with full bunkers the effect of filling No. 2 deep tank forward (420 tons S.W.) is to increase the draught forward by about 3 ft., 6 in., but reduce propeller immersion by 1 ft., 4½ in. Thus, in the case of Condition III, at the end of a short voyage of 6000 miles with draughts 5 ft., 8 in. forward and 17 ft., 1½ in. aft, and assuming that for reasonable behavior in head seas the draught forward should be at least 8 ft., then the most that could be expected aft would be about 16 ft., obtained by partial filling of forward deep tanks. This would leave 3 ft., 9 in. of the blade tips breaking surface, or nearly half the effective blade length.

If bunkers were consumed from No. 3 deep tank instead of Nos. 2 and 3 double bottoms, as in Condition IV, propeller immersion would suffer even more, with a draught aft of 14 ft., 4 in. only. This procedure, fortunately, is unlikely to be followed in practice.

Conditions V, VI, and VII are intended to represent the condition with fuel almost burnt out at the end of a very long voyage (Condition V), and to show the effects of filling water ballast, first in No. 3 deep tank to improve propeller immersion (Condition VI) and, secondly, of adding to this water ballast in No. 2 deep tank to correct draught forward to at least 8 ft. (Condition VII). It is significant that under this latter condition a draught aft of only 14 ft., 7 in. can be expected, leaving 5 ft., 2 in. of the blade tips breaking surface.

For comparison with this latter condition a similar calculation has been made for the "Ocean" type, Condition III (end of long voyage), from which it will be seen that with about the same draught forward (8 ft., 1½ in.) more than 2 ft. better immersion is obtained aft; in fact, with this class of coal-burner there seems little need to go much below 17 ft. aft.

The comparison between the two types would clearly show even more to the advantage of the "Oceans," where, as is probably more common from considerations of the price and availability of steam coal, bunkering is carried out in U.K. ports, often before making a westward North Atlantic passage in ballast, instead of at the loading port as probably has been more usual practice with the "Libertys."

It is well known that the conditions of war service demanded deeper ballasting of tramp type steamers, partly on account of the routing of convoys in higher latitudes of the North Atlantic. As Sir Amos Ayre has stated,* in some cases this was provided in the form of additional dry ballast, but in later vessels of the British wartime program permanent water ballast tanks were added of approximately 1000 tons capacity distributed abreast the tunnel, in the engine-room at the wings, and at each side—port and starboard—of the fore-hold bulkhead, thus ensuring the minimum loss of useful cargo space. In many cases, however, considerable dry ballast was carried in addition to this extra 1000 tons.

That a similar problem existed with the "Libertys" is evident from the fact that during the war the U.S. Maritime Commission introduced a special scheme of ballasting in which 1500 tons of dry ballast was carried in Nos. 2, 3, and 4 holds, although this was partly with a view to reducing hull stresses. Conditions VIII and IX, Table VI, show the disposition of the ballast under this scheme, the latter indicating the approximate condition after a North Atlantic return voyage with bunkers previously taken in North America sufficient

* Ayre, A. L. 1945. Trans.I.N.A., Vol. 87, p. 8, "Merchant Shipbuilding during the War."

for a 6000-mile voyage (plus margin) only. It will be seen that by this means a favorable comparison with the "Ocean" class was obtained. Naturally, under peace-time conditions such a scheme would generally be impracticable, and it is understood few, if any, of these vessels carried this additional ballast over the period under review. The statistics included in Table I lend support to this statement.

One of the difficulties with the "Liberty" type appears to be that the capacity of the forward deep tanks cannot be fully employed to correct for the consumption of fuel forward and provide adequate trim for good steering and avoidance of pounding without at the same time appreciably reducing the propeller immersion.

It may here be recalled that Sir Amos Ayre * has recommended that this class of vessel as suitable for peace-time service (except in the north-North Atlantic) a mean ballast draught of 16 ft., 5½ in. when fully bunkered (draught forward 12 ft., 4 in., aft 20 ft., 7 in.). This would approximate fairly well to "Liberty," Condition VIII, or "Ocean," Condition I of Table VI. H. Bocler † has also made tentative recommendations for minimum mean draughts in ballast based on mean depth allowing for side area of superstructure. From ballast draughts quoted in his paper for typical vessels of 400 to 450 ft. in length it would seem that with a small quantity of fuel remaining and propeller 80 percent immersed a mean draught of about 0.33 of mean depth would be representative of prewar practice. On this basis for the "Liberty" vessel minimum arrival draughts of 16 ft. aft and 9 ft., 2 in. forward (mean 12 ft, 7 in.) would seem appropriate, i.e., approximating to "Ocean," Condition III, but with less trim by the stern.

DISCUSSION OF AVAILABLE EVIDENCE

It is known that during the late war the "Liberty" and similar vessels were operated mainly in 9-knot convoys with engine speeds averaging about 60 Rpm., that is, well below the 3rd order torsional critical at 76 Rpm. This, together with the special precautions adopted in ballasting, probably protected the machinery against heavy racing. Since the war, however, most vessels have been operating with average engine speeds between 68 and 74 Rpm., possibly with occasional short bursts of higher speeds. Furthermore, with the virtual abandonment of the war-time practice of carrying additional dry ballast, the arrival draughts of very many of these vessels have left a great deal to be desired (see Tables I and VI). The author has spoken to some masters and engineers, and it seems generally agreed that when in ballast these vessels can race very heavily. Furthermore, a good many of the engines were not originally fitted with governors and with, possibly in some cases, less experienced crews, speed control by hand throttle may not always have been too prompt.

At this stage it may be well to recall that in many instances (see Figs. 10 and 11) the appearance of the fracture has exhibited the well-known "oyster shell" or "tree ring" markings so characteristic of many fatigue failures. It is known that these markings are symptomatic of periods of different rates of propagation of the fatigue crack depending upon the intensity of overstress applied. These are the very conditions which characterize the service of a "Liberty" type screwshaft, i.e., occasional periods of heavy overstress during racing, fol-

* Op.cit.

† Bocler, H. 1941-2. Trans.N.E.C.Inst. Eng. and Shipbuilders, Vol. 58, p. 241, "The Ballasting of Cargo Ships."

lowed by longer periods of continuous operation at moderate fatigue stresses in the vicinity of the service speed.

The latter stresses, i.e., from ± 3000 to ± 4000 lb. per sq. in. in torsion, and say ± 4000 lb. per sq. in. in reversed bending, are not considered to constitute by themselves the prime cause of failure. Bearing in mind, however, the disastrously weakening effect of leaky rubber sealing rings and the possible stress concentrations caused by faulty fitting or machining of the keyed assembly, particularly when associated with the seriously damaging effects of previous heavy overstress during racing, it is considered that as a precautionary measure continuous operation in the vicinity of the 3rd order 1-node torsional critical, coincident with the designed maximum service speed of 76 Rpm., should be avoided (see Appendix V).

This opinion is held despite the good record of the "Ocean" class, which have successfully withstood vibration stresses of equal magnitude at the same service speed, because in the "Liberty" vessels damage may well have been done already to the existing screwshafts by previous overstress during racing. It is important to note that such damage can exist without the actual initiation of cracks.

As regards the serious consequences of racing through the 2nd order critical at 120 Rpm., with stresses of ± 10,000 lb. per sq. in. and upwards in the screwshaft, it has been asked why more cases of shrinkage fit failures in the after L.P. crank webs have not been experienced (American sources quote only nine failures, of which four were associated with loss of propeller at sea), and reference has been made to the results reported by Forsyth* for a similar but smaller installation (without governor) in which the flank of the 2nd order critical was picked up at more than double the normal service speed of 70 Rpm., giving rise to a stress of at least ± 6000 lb. per sq. in. in the screwshaft superimposed on estimated occasional maximum stresses due to acceleration of about 2200 lb. per sq. in. In this classic case it was stated that failure of the shrinkage fit in the after L.P. crank web occurred on six occasions within 4½ years. The trouble with this vessel was cured by increasing the shrinkage area of the after L.P. crank web and fitting a suitable governor.

As a matter of interest it has been thought well to make calculations on the relative strengths of the shrinkage fits of the after L.P. crank webs in the two installations, neglecting shear strength of the dowel pins (fitted in both installations). For this purpose a shrinkage allowance of 1/600 of the shaft diameter in the eye-hole has been assumed for each case and a value of 0.25 for the coefficient of friction. The latter figure may possibly be conservative, since R. Russell† has shown that for a small (and thus presumably dimensionally accurate) mild steel shrunk assembly with an allowance of only 1/1500 of the diameter, the value of the coefficient of friction in torsion amounted to 0.29 and 0.9 when assembly was made with a rape oil film and perfectly dry, respectively.

Expressing the torsional strength of the shrunk web assembly at slip in terms of corresponding shear stress in the screwshafts of the two installations, and with the above-mentioned assumption (neglecting the shear strength of the dowel pins), the following results have been calculated, applying the usual Lamé formula for the radial grip pressure:—

* Forsyth, G. H. 1943. Trans.I.Mar.E., Vol. 55, p. 101, "The importance of governing arrangements for Marine Installations with Special Reference to Torsional Vibration."

† Russell, R. 1933. Proc. I.Mech.E., Vol. 125, p. 493, "Factors affecting the Gripping Force, Shrink and Expansion Bits."

Forsyth's vessel—14,000 lb. per sq. in. in 12-inch diameter screwshaft.

"Liberty" vessel—23,500 lb. per sq. in. in 15½-inch diameter screwshaft.

The principal factors affecting these results are the relatively large diameter screwshaft in the former vessel, and the fact that in the "Liberty" crankshaft the diameter of the eye-hole in the web is increased by 1 inch over that of the journal and a correspondingly greater axial length provided.

From the foregoing it can be concluded that the almost complete immunity from crankshaft failures in the "Liberty" vessels by no means proves that their screwshafts have not been subjected to severe torsional stresses.

CONCLUSIONS

It is believed that the many failures in the screwshafts of vessels of the "Liberty" and similar types are primarily the result of insufficient propeller immersion in the ballast condition leading to excessive and frequent racing, which in turn has subjected the shaft to heavy vibration stresses, both torsional and bending. These stresses, taken in conjunction with existing stress raisers, have either been sufficient to cause early cracking with resulting rapid failure, or alternatively, have so weakened the shaft that fatigue failure has been brought about more slowly through the alternate action of overstress and the moderate vibration stresses occurring near the service speed. Contributory factors have been the effects of corrosion fatigue from leaky shaft seals, slack-fitting propellers and keys, and unnecessarily severe stress concentrations.

In the light of the foregoing it is difficult to avoid the conclusion that serious consideration should now be given to the provision of additional water ballast capacity in oil-burning ships of the "Liberty" and similar types. Where hull strength is adequate, a practical solution could be obtained by the fitting of additional tanks abreast the shaft tunnel in No. 5 hold and the resulting loss of cargo space could be offset to some extent by reducing the depth of the forward deep tanks, which, in any case, at the present time are largely ineffective in improving the vessel's immersion.

It is believed that where steps are taken to ensure propeller immersions comparable with those usual in the coal-burning "Ocean" class, together with precautions, as indicated, to avoid continuous operation at or near the existing 3rd order critical speed, these measures will confer adequate protection against further screwshaft failures in this class of vessel, particularly where attention has been paid to the many smaller, but nevertheless important, details, of which examples have been given in the paper.

ACKNOWLEDGMENTS

The author's thanks are due to the Committee of Lloyd's Register of Shipping for permission to publish this paper, to Dr. S. F. Dorsey, C.B.E., F.R.S., and Mr. W. J. Ferguson, M.Eng., for their encouragement and support; further to his many colleagues both at home and abroad who gave information and help, in particular to those in the Engineering and Ship Research Departments, and to Mr. B. Moor for his staunch assistance in the work of compilation.

Grateful acknowledgment is also made to the following among others:—

The American Bureau of Shipping.
Cayzer, Irvine and Co., Ltd.
Wm. Thomson and Co. (Leith).
Silver Line Ltd.
Alfred Holt and Co.
J. and C. Harrison, Ltd.
Park Steamship Co. (Montreal).
The London Salvage Association.
Lloyd's Shipping List.
Meteorological Office, M.O.1, Air Ministry.

List Of Appendices

(Refer to original article for full appendices)

Appendix I—The characteristics of the 1-node mode of damped torsional vibration of the engine-shafting propeller system with particular reference to the interpretation of seismic type torsiograph records.

Appendix II—Calculations for determining torsional vibration stresses from torsiograph records (Fig. 14).

Appendix III—General calculations on torsional vibration.

Appendix IV—Calculations on bending fatigue.

Appendix V—Precautions against the damaging effects of torsional and flexural vibration.

BOOK REVIEW

"STEAM TURBINES AND THEIR CYCLES"

BY J. KENNETH SALISBURY

REVIEWED BY RUDOLPH MICHEL,
Member, A.S.N.E.

Publisher: JOHN WILEY & SONS, INC., New York, N. Y. Price: $9.00. CHAPMAN AND HALL, LTD., London.

This book is a most valuable contribution to steam turbine and power plant literature. The book is divided into four parts, covering (a) thermodynamic and turbine fundamentals, (b) the regenerative cycle, (c) cycle analysis and (d) application engineering.

The review of fundamentals presented in Part I of the book is most refreshing in the manner of presentation. Much valuable turbine design data not previously covered in American texts is presented. Not only does the author present design practice of his own company, but the published methods of other authoritative sources are presented.

Part II gives a comprehensive treatment of heat balance calculations for ideal cycles. Derivations of formulæ are given as well as examples of their use. The short-cut method for evaluating heat rates of regenerative cycles, originated by the author, is of particular value.

In Part III the author presents his method, of the analysis of actual power plant cycles. The method is unique in that it provides a means of determining small deviations, in an actual cycle, of the heat supplied and rejected in the ideal cycle. These deviations are a measure of the change in heat rate from the ideal. Differences in plant heat rate, resulting from changes in feed water heating arrangements, are obtained by direct calculation of the differences rather than by separate calculation of two heat rates and determining the difference by subtraction.

Part IV deals with practical design details and estimation of turbine steam rates and heat rates. In the Appendix a valuable set of steam table data is given. The enlarged Mollier diagram, printed in sections on separate pages, is of particular value to turbine designers.

One wonders why such important factors as stress calculations of the various turbine parts were omitted. Also, there is no discussion of the problems of vibration and critical speeds usually encountered.

The language of the book is unusually clear and direct. In addition, the excellent format, printing and many curves and diagrams add to its usefulness. It is probably the most important American text published on the subject.

"THE JAPAN SCIENCE REVIEW"

We have just received Volume I, Number 1, of The Japan Science Review, Series I, Engineering Sciences. It is to be published quarterly by Association for Science Documents Information, Tokyo Institute of Technology. Subscription rate is $5.00 per year. Subscriptions should be mailed to

THE JAPAN SCIENCE REVIEW
ENGINEERING SCIENCES
Executive Office: Association for Science Information,
% Tokyo Institute of Technology,
Ookayama, Meguro-Ku,
Tokyo, Japan

The first issue is of 212 pages printed by the offset process. We are advised that plans include improvement in the method of printing. Even as printed the "Review" is clear and easy to read. Some of the halftones are not as clear as might be desired.

The table of contents lists 31 titles covering a very broad segment of the field of Engineering Sciences. It will be a valuable addition to an engineering or scientific technical library.

The entire "Review" is written in English.

CHANGES IN MEMBERSHIP

This section will, starting with this issue, be used to permit the membership list, which was issued as of 1 May 1950, to be kept current.

OMISSION

The Secretary regrets that the name of
Lieutenant Commander W. J. McDonald, U.S.N.R.
c/o Downtown Athletic Club, 19 West St.
New York, N. Y.
who has been a naval member, was omitted from the published list.

ERRORS

It is also regretted that the following errors were not corrected.
On page 59, Naval Member
John G. Wilson was printed as John B. Wilson, and
On page 70, Civil Member
Richard W. Frost was printed as Robert B. Frost.

It will be greatly appreciated if any other omissions or errors are brought to the attention of the Secretary.

NEW MEMBERS

The Society is pleased to announce the acceptance of the following new members since the publication of the Membership List as of 1 May 1950.

NAVAL

Brown, James Andrew, Commander, U.S.N.,
1613 N. Edgewood St., Arlington, Va.

Carlson, Sidney Oscar, Jr., Lieutenant, U.S.N.R.,
Instructor, U.S. Marine Academy, Kings Point, N. Y.

Crowley, Charles William, Lieut. Comdr., U.S.N.R.,
Asst. Div. Mgr., Opr., The Texas Company
Mail: 1200 East 95th St., Seattle 9, Wash.

Davis, William Howden, Lieut. Comdr., U.S.N.R.,
Asst. to Vice President, Alloy Engrg. & Casting Co., Champaign, Ill.

DuBon, Frederick Valentine, Jr., Ensign, U.S.N.,
U.S.S. *Roanoke* (CL 145), c/o Fleet P. O., New York, N. Y.

Johnson, Julian Frank, Lieut., j.g., U.S.N.R.
Graduate Student, Brown University
Mail: Metcalf Research Laboratory, Brown University, Providence 12, R.I.

Kauffman, Hal A., Lieutenant, U.S.N.,
1924 5th St., Bremerton, Wash.

Krepps, Phillip Calvin, Lieutenant, U.S.N.R.
Turn Foreman, Power & Fuel, Upper Div. Nat. Tube Co. National Works, McKeesport, Pa.
Mail: 2914 John St., Homestead Park, Pa.

Liebrecht, Lawrence Clifford, Lieutenant, U.S.N.R.,
Shady Lane, Mundelain, Ill.

McCaffery, Elmer Vernon, Chf. War. Mach., U.S.N., Ret.
Field Engineer, Green Fuel Economizer Co., Beacon, N. Y.
Mail: Upper Station Rd., RFD, Garrison, N. Y.

McElroy, John Charles, Ensign, U.S.N.,
U.S.S. *Wright* (CVL 49) c/o Fleet P. O., New York, N. Y.

Morse, Duane Edward, Chief Radio Elect., U.S.N.,
U.S.S. E-LSM 445, c/o Fleet P. O., New York, N. Y.

Puidokas, Stanley Victor, Lieutenant, j.g., U.S.N.R.
Asst. Development Engineer, Nash Motors Div., Kenosha, Wis.
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Smiles, Abner Sherman, Lieutenant, j.g., U.S.N.R.
Webb Institute of Naval Architecture
Crescent Beach Road, Glen Cove, L. I., N. Y.

Stello, Karl Henry, Lieutenant, j.g., U.S.N.R.
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U.S. Dept. of State International Broadcast Div., Honolulu
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Watts, Charles Robert, Captain, U.S.N.,
Code 550, Bureau of Ships, Navy Dept., Washington, D. C.

CIVIL

Barnard, Clayton, Application Engineer, Bailey Meter Co.,
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Israel-America Line, 27 William St., New York, N. Y.
Mail: Balforia, Post Affula, Israel

Eaton, Reginald Orbin, Commander, U.S.N.R.,
Naval Service Engineer, General Electric Co., Los Angeles, Calif.
Mail: 1864 Chatwin Ave., Long Beach 4, Calif.

Gerrick, H. Georges, President, Randolph Engineering Co.,
2300 First National Bank Building, Pittsburgh 22, Pa.

Gordon, Britton L., President, Blackmer Pump Co.,
Grand Rapids, Mich.
Mail: 1809 Century Ave., SW., Grand Rapids, Mich.

Harris, Henry Hickman, President, General Alloys Co.,
Boston, Mass.
Director U.S. Naval Research & Development Cont. No. Nobs -45472 at
Alloy Engineering & Casting Co., Champaign, Ill.

Lubinski, Robert Albert, Planner & Estimator,
New York Naval Shipyard
Mail: 471 17th St., Brooklyn 15, N. Y.

Luke, Roger M., Asst. to Chief Engineer, Hyde Windlass Co.,
Mail: 103 Middle St., Bath, Maine

Mullestein, William E., Manager Field Sales,
Lukens Steel Co., Coatesville, Pa.

Phillips, Henry Weldon, Project Staff,
General Electric-Knolls Atomic Power Laboratory, Schenectady, N. Y.

Taylor, Warren F., Mech. Designer, Pioneer Service & Engineering Co.,
231 So. LaSalle St., Chicago, Ill.

Worth, John Sharpless, Asst. Metallurgical Engineer,
Bethlehem Steel Co., Bethlehem, Pa.

ASSOCIATE

Fitzmaurice, Edmund Joseph, Jr., Sales Engineer,
Electric Storage Battery Co., Washington, D. C.
Mail: 4409 Warren St., Silver Spring, Md.

Halpern, Benjamin, Chf. Engineer of Government of Israel Navy Dept.
Mail: 2288 Mott Ave., Far Rockaway, L. I., N. Y.

Marbury, Fendall, Jr.,
234 West Lafayette Ave., Baltimore 17, Md.

Nelson, Joseph Frances, Lieutenant, j.g., U.S.M.S.
3d Engr., United Fruit Co.
Mail: 8835 Floyd St., Jacksonville, Fla.

Roosa, Herbert H., Vice President, Munzel, Inc.,
315 Babcock St., Buffalo 10, N. Y.
Mail: 54 Sargent Drive, Snyder 21, N. Y.

The following have resigned:

NAVAL

Cator, George A.
Farrel, Walter J.
Stott, A. C.

CIVIL

Dilloway, T. M.
Gibson, W. R.

The Society has learned of and announces with deep regret, the deaths of

WILLIAM RICE BALLOU, Naval Member

and

WARREN NOBLE, Naval Member

ASSOCIATION NOTES

Original Articles

The editor of the JOURNAL is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $40.00 to $150.00 depending upon length, interest and professional value.

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. Include on a separate page a short biographical sketch(es) of the author(s). 50 to 100 words for each author is desired.

Manuscripts should be mailed addressed to:

Secretary-Treasurer
American Society of Naval Engineers
605 F St., N. W.
Washington 4, D. C.

Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 10 reprint copies of his article free of charge. Additional copies may be purchased from the Society if the request is received 30 days prior to the publication date which is the 25th of the issue month. Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Manuscripts which are accepted will be published in the earliest JOURNAL which has not yet been closed (60 days before publication) and for which insufficient material is already on hand.

Subject Matter and Authors

To assist the editor in programming future issues, the help of all members is solicited. To make it easy, the last page in this issue is readily detachable as you will find. This page has blanks for two entries.

(*a*) What subject would you, as a member, like to see covered by an article in the JOURNAL.

and

(*b*) Who do you consider best qualified to prepare such an article authoritatively, and what is his address. (if you know).

Any of these forms which are filled in and mailed to the Society will be received with thanks and the editor will follow up to get the suggested articles prepared.

Photographs

Photographs of current or historical interest to readers of the JOURNAL are desired. Any which are accepted by the Society will be purchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

Notice of Deaths

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be appreciated greatly if any member who learns of the death of a member will advise the Secretary-Treasurer.

Society Lapel Button

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold back-ground.

It is available to all members at fifty cents (50c) each.

LIFE MEMBERSHIPS

Any member may now purchase a life membership under the following rules:

(*a*) Any dues paid for the year in which life membership is purchased will be credited to the cost of the life membership.

(*b*) Life memberships are non-transferable and terminate with the death of the member.

(*c*) No refund will be made on account of death or resignation of a life member.

(*d*) Life membership will vary in cost with the age of the member at his next birthday following application, as follows:

Over 55 years	$125.00
40 to 55 years	175.00
25 to 39 years	225.00
Under 25 years	250.00

(*e*) All life memberships paid in shall be protected by a transfer on the books of the Society of Series G, U. S. Government 2½ per cent bonds. Thus a life membership fund will be established. It will be credited with moneys paid in for life memberships and debited with the regular amount of annual dues for each life member each year.

LOCAL CHAPTERS

Local Chapters of the American Society of Naval Engineers may now be authorized by the Council. The following rules for recognition of Local Chapters have been adopted by the Council:

(*a*) Twenty or more regular members, Naval or Civilian, may apply for a Charter as a Local Chapter.

(*b*) Each application must be accompanied by a list of members and a copy of local By-laws. The latter must meet the following minimum requirements.

1. All financial dealings of the local Chapter must be independent of the Society and can in no way obligate the Society as a whole. The local chapter shall set up on a non-profit basis. All accounting shall be local.

2. Each active local Chapter shall receive from the Society a grant of $0.50 per year for each naval member, civil member or associate who resides in the local area and who associates himself with the local Chapter. This grant may be used by the local Chapter to defray any necessary expenses, including (a) Letterheads, (b) Notice cards, (c) Postage, (d) Minimum rental allowance, (e) Secretarial service (part time).

3. Any technical papers presented at local chapter meetings shall be submitted to the Secretary-Treasurer of the Society for consideration for publication in the JOURNAL. Any paper accepted will be paid for at regular rates.

4. Each local chapter shall submit a quarterly report of its activities to the Secretary-Treasurer. These reports must reach Washington, D. C. prior to 15 January, 15 April, 15 July and 15 October. This report shall contain a list of active local members as of the first day of the quarter for the purpose of computing the amount of grant.

5. Associate members of the Society shall be eligible for membership in a Local Chapter, entitled to all the privileges of other members except voting and holding office.

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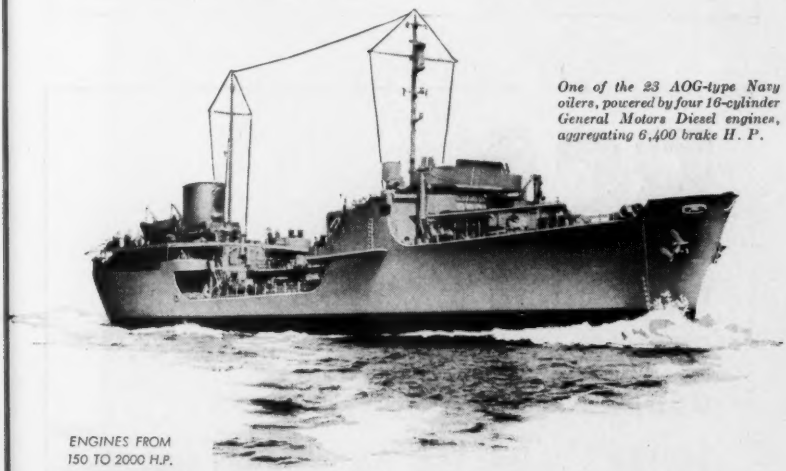

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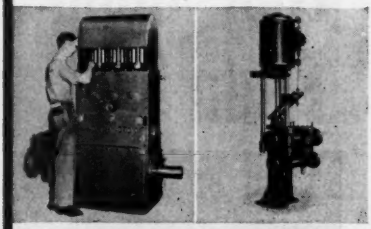

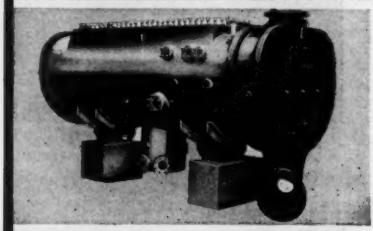

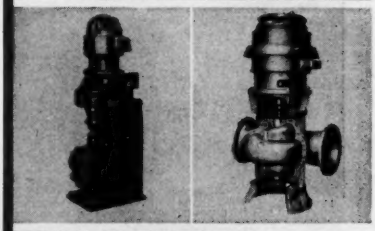

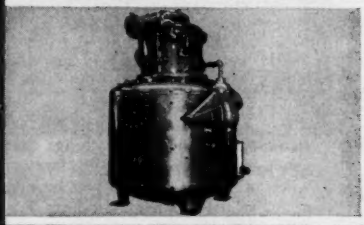

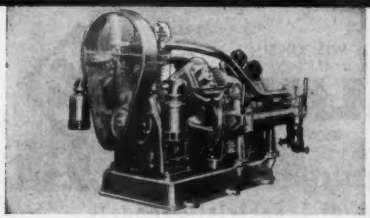

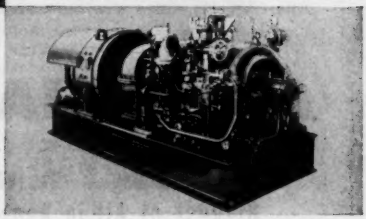

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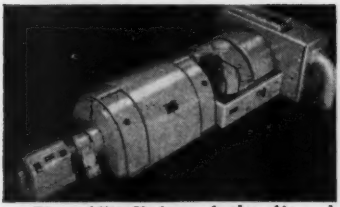

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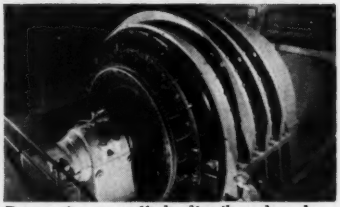

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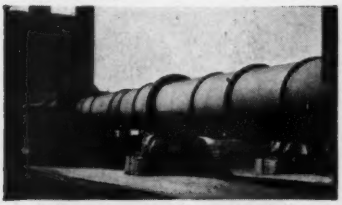

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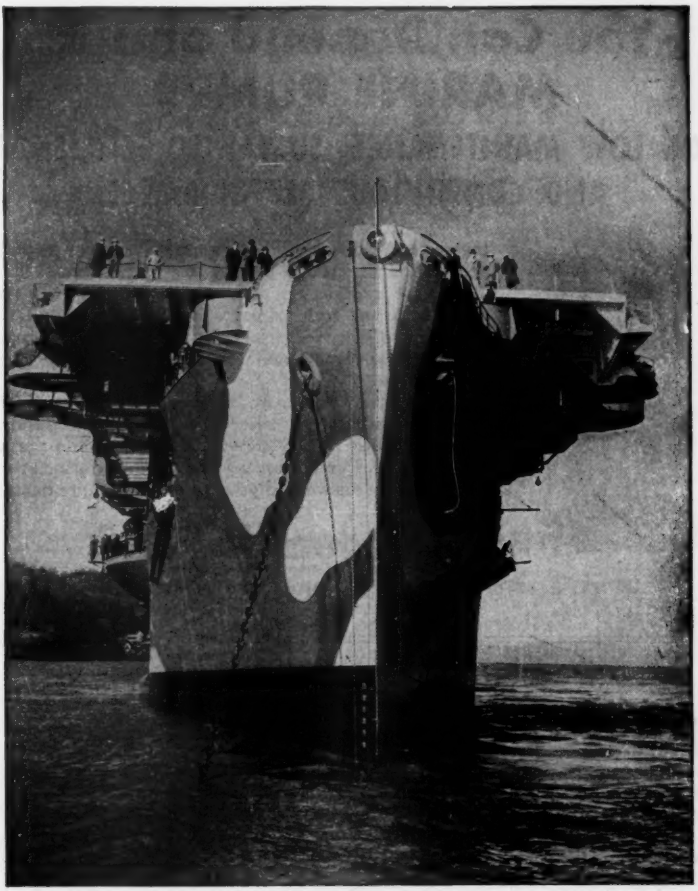

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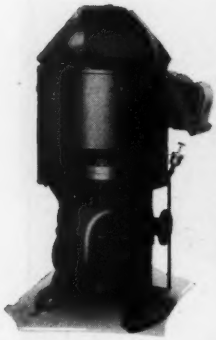

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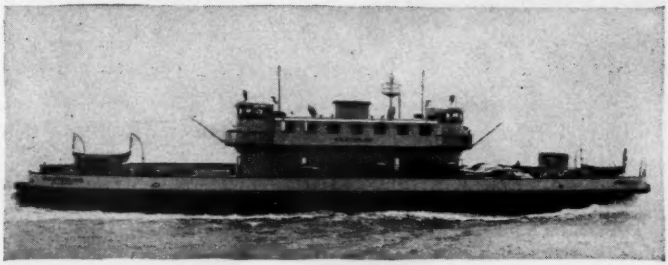

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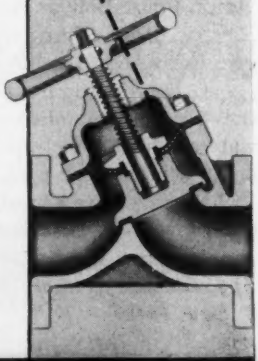

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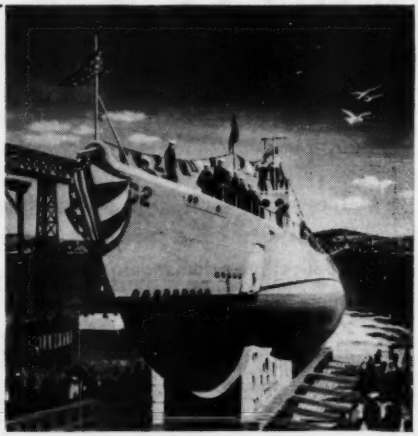

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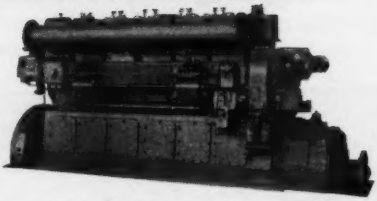

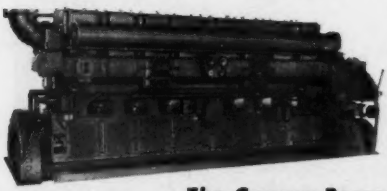

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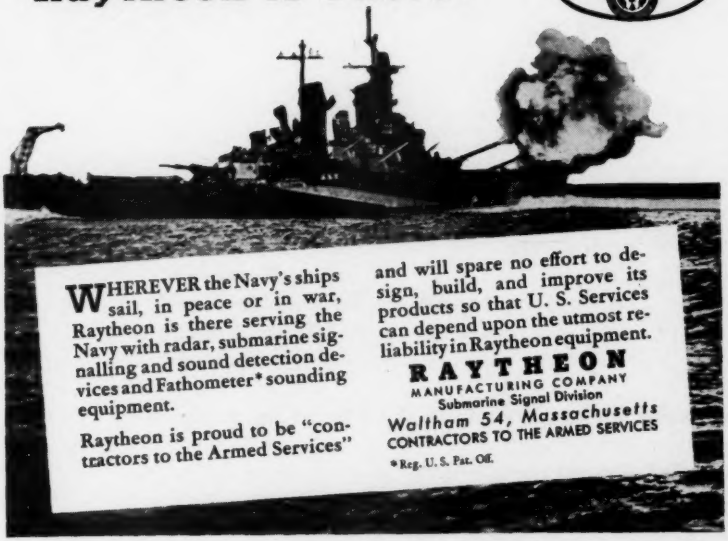

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